Design and Performance Evaluation of Evaporative Towers

by

Haitham M. S. Bahaidarah

A Thesis Presented to the

FACULTY OF THE COLLEGE OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

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DEANSHIP OF GRADUATE STUDIES

This thesis, written by **Haitham Muhammad Sadagah Bahaidarah** Under the direction of his Thesis Advisor, and approved by his Thesis committee, has been presented to and accepted by the Dean of Graduate Studies, in partial fulfillment of the requirements for the degree of **Master of Science in Mechanical Engineering.**

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Date: 7/6/99

Dedicated to

My Parents,

Brothers, Sisters,

Wife & daughter

Acknowledgment

In the name of Allah, Most Gracious, Most Merciful

'Of knowledge, it is only a little that is communicated to you' (Qur'an 17:85). All Praise to Almighty Allah, who gave me chance, patience, ability and time to accomplish this work.

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Last but not least, thanks are due to the member of my growing family for their emotional and moral support throughout my academic career. No personal development could ever take place without the proper guidance of parents. Thanks are also due to my brothers and sisters for their prayers and encouragement. My elder brother, Saeed, deserved special mention for his sacrifice and inspiration. Special thanks are also to my patient wife and loving daughter.

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Nomenclature

a	Surface area per unit heat-exchange volume (m^2/m^3)
A	Approach = T_{w2} - T_{wb} (^{O}C)
A	Surface area (m^2)
B_X	Bias error
C_0, C_1	Constant
C_p	Specific heat of fluid (J/kg ^{O}C)
C	Heat Capacity (J/ ^{O}C)
δ	A variation
ε	Thermal effectiveness
ε_{Xi}	Normalized uncertainty in parameter X_i, dimensionless
ε_Y	Normalized overall uncertainty in parameter Y, dimensionless
h_c	Convection heat transfer coefficient (W/m^2K)
I	Integral
i	Enthalpy of moist air (kJ/kg)
i_g	Enthalpy of saturated water vapor (kJ/kg)
K_m	Mass transfer coefficient (kg/m^2 s)
Le	Lewis number of moist air
M	Mass flux = $\dot{m}$/A (kg/m^2 s)
$\dot{m}$	Mass flow rate (kg/s)
n	Integer number
NU	Normalized uncertainty

NSC Normalized Sensitivity Coefficient

NTU Number of transfer units

P_x Precision error

Q Heat transfer rate (W)

R Range = $T_{w1} - T_{w2}$ (^{O}C)

RC Relative contribution

T Temperature (^{O}C)

U_{xi} Uncertainty in parameter X_i, unit of X_i

U_Y Uncertainty in parameter Y, unit of Y

V Volume of the tower (m^3)

W Humidity ratio (kg water per kg air)

X_i General input variable

X Nominal value of X, units of X

Y Nominal value of Y, units of Y

Z Heat exchanger depth in air flow direction (m)

Subscripts

1 Inlet condition

2 Outlet condition

a Air

i At air-water interface

W Water

Wb Wet bulb

ABSTRACT

Name: Haitham M. S. Bahaidarah

Title: Design and Performance Evaluation of Evaporative Cooling Towers

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Cooling towers are used to reject heat from water to the atmospheric air by means of heat and mass transfer. Since some of the water, which is in direct contact with air, is evaporated in the air the process is also known as evaporative cooling. The thermal analysis of wet cooling tower has received considerable attention in recent years. The effect of fill packing system on the performance of cooling tower has been the major consideration of many researches. In the present numerical study, the analysis of the thermal performance of cooling tower is carried out for different types of the fill packing available in the literature as well as from different manufacturers. Also, a comparison is made for the thermal characteristics of a cooling tower when a different type of fill packing system is used. In this regard, an experimental data for a small cooling tower having a capacity of 20 gal/min is used to validate the numerical results. It is found that the simulation results are within ± 1.2 % of the experimental values. In addition to the above investigation on fill packing system, a sensitivity analysis on the thermal characteristics of cooling tower is also presented with respect to the important parameters such as inlet water temperature, wet bulb temperature, tower characteristics, etc.

Master of Science Degree

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Dhahran, Saudi Arabia

May 1999

الخلاصة

الاسم : هيثم محمد صدقة باحيدرة

عنوان الدراسة : تصميم و تقييم أداء أبراج التبريد بالتبخير

المجال : الهندسة الميكانيكية

تاريخ الدرجة : مايو ١٩٩٩

تستخدم أبراج التبريد لطرد الحرارة من الماء إلى الغلاف الجوي بواسطة انتقال الحرارة وانتقـــال الكتلـــة ، ولأن بعـــض جزيئات الماء تتبخر عند اتصالها مع الهواء مباشرة فإن هذه العملية تعرف بالتبريد بالتبخير. لقد لاقى التحليـــل الحـــراري لأبراج التبريد اهتماما كبيرا من الباحثين في السنوات الحالية وخاصة تأثير الحشو أو نظـــام التعبئـــة علـــى أداء بـــرج التبريد.وفي هذه الدراسة الرقمية يدرس التحليل الحراري لأداء أبراج التبريد لنماذج مختلفة من الحشو أو التعبئة الموجـــودة في البحوث السابقة مع إجراء مقارنة للخصائص الحرارية لكل منها. لهذا الغرض استخدمت بعض المعلومات المعملية لبرج تبريد صغير سعته عشرين جالونا في الدقيقة لبرهنة النتائج الرقمية. وقد وجدت النتائج في نطاق ١,٢% مقارنـــة بـــالقيم المأخوذة من النتائج المعملية. بالإضافة إلى ما ذكر, درست مدى حساسية العوامل المؤثرة مثل درجة حرارة الماء الداخـــل ودرجة حرارة الهواء المحيط على الخصائص الحرارية لبرج التبريد.

درجة الماجستير في الهندسة

جامعة الملك فهد للبترول و المعادن

الظهران – المملكة العربية السعودية

CHAPTER 1

INTRODUCTION AND BACKGROUND

When water changes its state from liquid to vapor or steam an input of heat energy must take place, which is known as the latent heat of evaporation. This input energy must either be supplied from fuel as in a boiler or be extracted from the surroundings. Cooling towers take advantage of this change of state by creating conditions in which hot water evaporates in the presence of moving air. By this means heat is extracted from the water and transferred to the air and the process is known as evaporative cooling. The principle is very simple but the heat transfer processes are quite complex. Typical cooling towers consist of no more than a four-sided wooden structure in which the hot water is introduced as a spray at the top of the tower, mixed with the cooling air and drawn off from a sump at the bottom. The water is thus cooled for return to the condenser or process to be cooled.

The principal criteria on which the design and manufacture of cooling towers is based are (Hill et al., 1990):

- Achieving maximum contact between air and water in the tower by the optimum design of tower packing and water distribution system.
- Assisting the flow of air by means of fans.
- Minimizing the loss caused by water spray escaping from the tower. It should be emphasized that control of spray loss is also of great importance in eliminating the risk of infectious diseases being transmitted to people by the warm moist air.
- Relating the design of the tower to the volume flow rate of the water to be cooled and to the three critical temperatures, i.e. ambient air wet bulb temperature, warm water inlet and cooled water outlet temperature.
- Assuming that problems arising from the quality of the water such as corrosion, fouling and the growth of bacteria are properly understood and controlled.
- Taking due account of space limitations at the tower's location and of the possibility that noise from the tower may be a source of nuisance to those living or working in the vicinity.

1.1 Main components

Figure 1.1 shows a schematic arrangement of a mechanical draught cooling tower, which consist of the following components:

<u>Casing or shell</u>: The structure enclosing the heat transfer process is properly reinforced to carry the other main items.

<u>Air inlet and air outlet</u>: The positions at which cool air enters and hot air leaves the tower. In natural draught towers the inlet is normally protected by drip-proof louvers and the

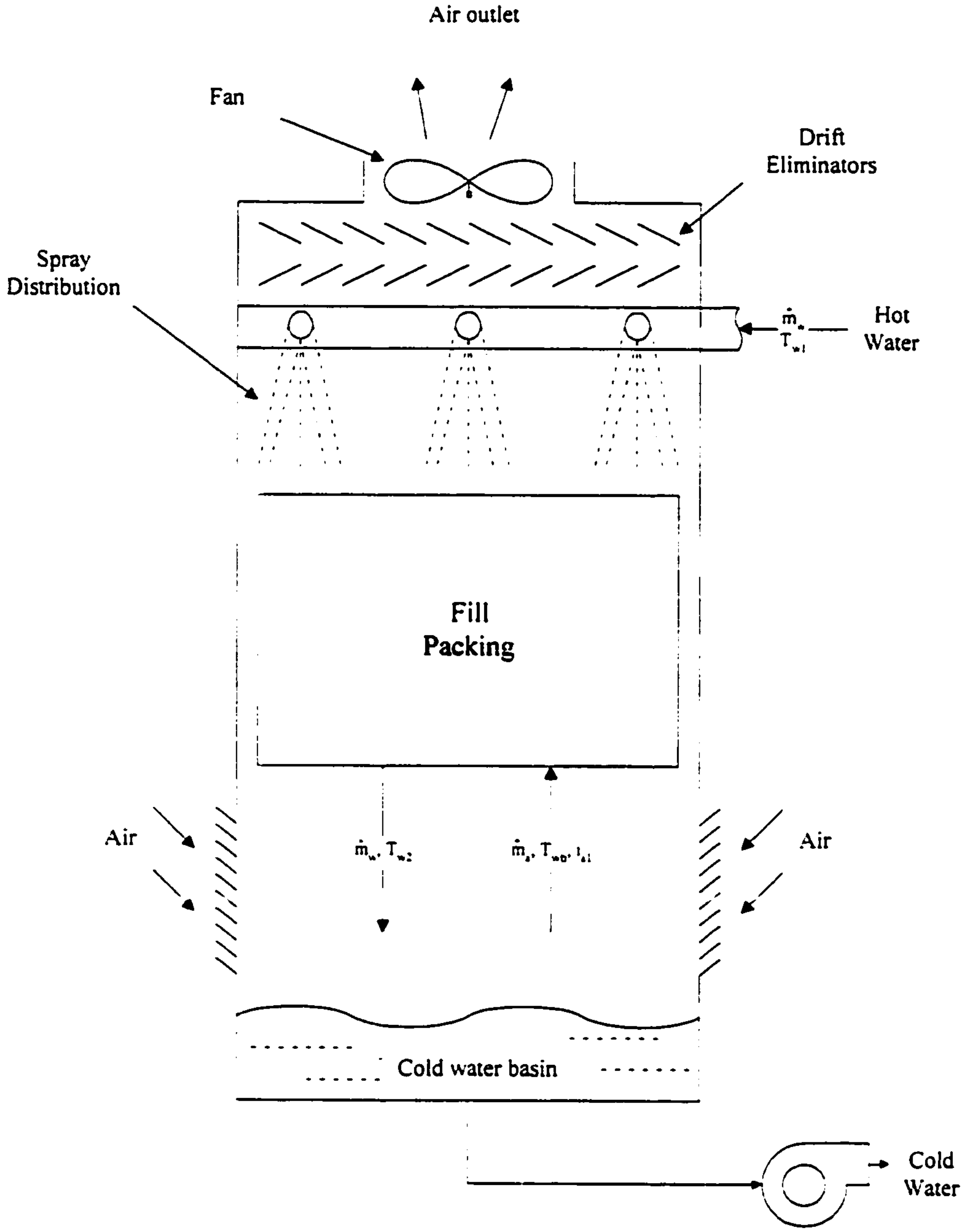

Figure 1.1: Schematic arrangement of a typical mechanical draught cooling tower.

outlet by a suitable grill. Where an induced draught fan is used the outlet is the fan casing; with forced draught the fan casing provides the inlet.

Fan: Correct selection of fan according to the tower duty is very important; volumetric airflow rate, fan pressure developed and noise from motor and fan impeller must all be considered according to the duty and location of the tower.

Drift eliminators: These are positioned in the outlet air stream so as to prevent water droplets from being carried away from the tower by the air stream.

Warm water inlet: The point at which the warm water from the process enters the tower.

Water distribution system: Water entering the tower must be spread as evenly as possible over the cross-section of the tower.

Packing: It is often described as a fill material. It consists essentially of a system of baffles which slows the progress of the warm water through the tower and ensures maximum contact between water droplets and cooling air by maximizing surface area and minimizing water film thickness. There are many different types of packing material that are used in the cooling tower.

Cold water basin: It is also referred to as tank or sump. The point at which the cooled water is collected before return to the process.

Cold water outlet: The point at which the cooled water leaves the tower.

1.2 The physical mechanisms of cooling tower operation

Basic mechanisms by which the water is cooled, are best described by Figure 1.2, which illustrates a single droplet of water in the tower. The droplet is surrounded by a thin film of air which is saturated and remains almost undisturbed by passing the air stream. It is

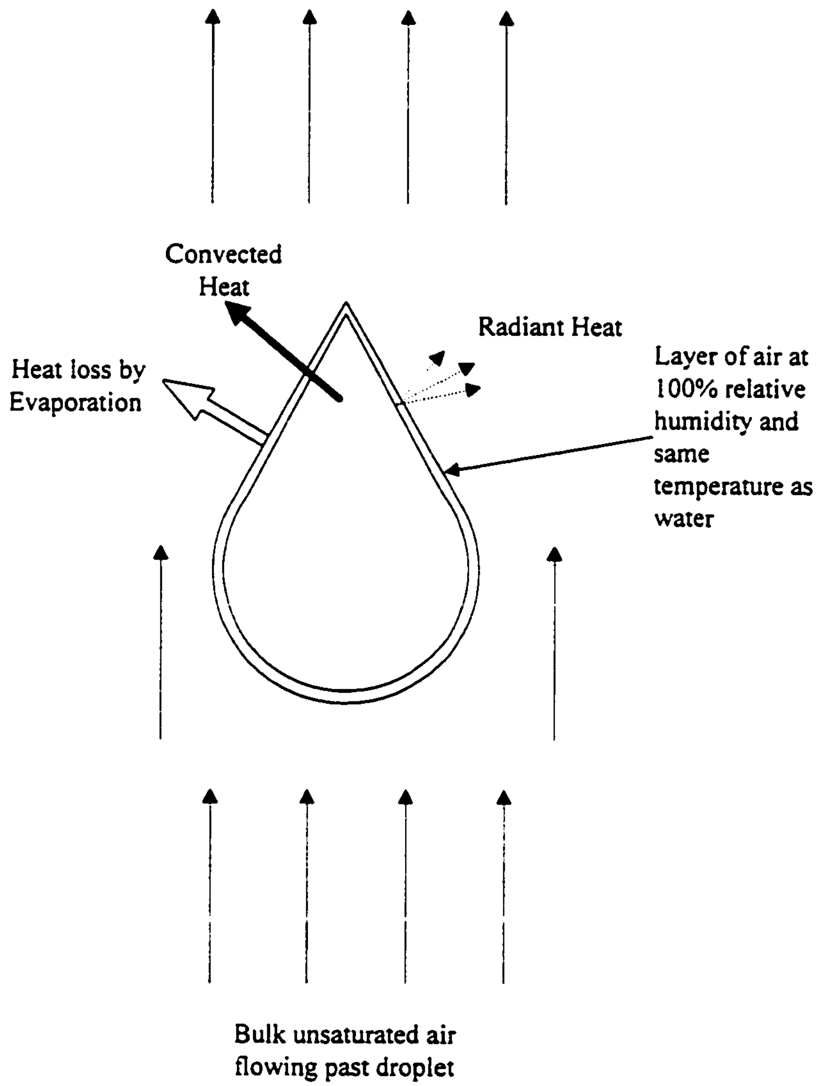

Figure 1.2: Diagram showing the various ways in which a water droplet loses heat (Hill et al., 1990).

through this static film of saturated air that the transfer of heat takes place in three ways, i.e. (Hill et al., 1990):

- By radiation from the surface of the droplet; this is very small portion of the total amount of heat flow and it is usually neglected.
- By conduction and convection between water and air; the amount of heat transferred will depend on the temperatures of air and water. It is a significant portion of the whole, and may be as much as one-quarter to one-third of the total heat flow.
- By evaporation; this accounts for the majority of heat transfer and is the reason why the whole process is termed ''Evaporative Cooling''.

As discussed above, evaporation is the key to the successful operation of cooling towers. The evaporation that occurs when air and water are in contact is caused by the difference in pressure of water vapor at the surface of water and in air. These vapor pressures are functions of water temperature and the degree of saturation of air, respectively.

In a cooling tower the water and air streams are generally counter-current so that cooled water leaving the bottom of the pack is in contact with the entering air. Similarly, hot water entering the pack will be in contact with warm air leaving the pack. Evaporation will take place throughout the pack. It should be noted that at the top of the pack, the fact that the air is nearly saturated, is compensated for by the high water temperature and consequently high vapor pressure. The amount of evaporation which takes place depends on a number of factors, including the total surface area the water presents to the air (which is why the fill pack design is very important) and the amount of air flowing. The greater the air flow, the more cooling is achieved. This is because of the fact that as the air flow rate increases, the rate of heat transfer increases. It is also very important to mention that the wet bulb temperature of the entering air has a very important effect. A lower wet bulb temperature produces a lower water outlet temperature.

The factors, which influence the performance of a cooling tower, may be summarized as follows (Hill et al., 1990):

1. The cooling range.
2. The approach.
3. The ambient air wet bulb temperature.
4. The flow of water to be cooled (or circulation rate).
5. The rate at which air is passed over the water.
6. The temperature level.
7. The performance coefficients of the packing to be used.
8. The volume of packing (i.e. height multiplied by horizontal cross-sectional area).

The overall objective of this thesis is to study the thermal performance characteristics of evaporative cooling towers. In the present study, an investigation has been carried out on different types of the fill packing available in the literature as well as from different manufacturers. Also, a comparison has been made for the thermal characteristics of a cooling tower when a different type of fill packing system is used. In addition to investigation on fill packing material, a sensitivity analysis on the thermal characteristics of cooling tower is also studied with respect to the important parameters such as inlet water temperature, wet bulb temperature, and mass flow rates of air and water.

An extensive literature review on the thermal performance of evaporative cooling towers is given in Chapter 2. It also includes discussion on the previous work related to the assumptions inherent in the calculation procedure of the cooling tower performance. Chapter 3 covers the different ways, which are used to characterize the types of cooling towers. Also, it includes the types of the cooling tower fills and discussed the ways in which it can be characterized.

Chapter 4 describes the mathematical formulation of the calculation procedure of the cooling tower performance. It describes the assumptions applied in this study, and the algorithm followed to perform the calculation procedure. Also, it summarizes the sensitivity analysis on the thermal characteristics of cooling tower that is carried out for the different types of cooling tower fills with respect to the important parameters such as inlet water temperature, wet bulb temperature, and mass flow rates of air and water. Finally, the methodology adopted in this study and the results obtained are presented in Chapter 5 followed by conclusion and recommendations in Chapter 6.

CHAPTER 2

LITERATURE REVIEW

The theoretical analysis of wet cooling towers has a long history, which has led to an excessively large number of publications. A complete review of the origin and early history of technical papers dealing with cooling tower theories is surveyed by Baker (1984). The main objective of early investigators of cooling tower is the proper understanding of heat and mass transfer mechanisms that are associated with cooling tower operation. Baker (1984) reevaluated different suggestions of coupling both the heat and mass transfer in a single driving force. He reported that Coffey and Horne (1914) proposed and proved that cooling performance depends on the wet-bulb temperature of the ambient air which is the lower limit of cooling. They obtained a single driving force based on vapor pressure at the wet-bulb temperature. On the other hand, Walker et al. (1937) used the air humidity as the sole driving force for the cooling process in cooling towers. London et al. (1940) used the enthalpy of the humid air-water vapor mixture as the actual driving force without explaining the details of the derivation. They also recognized, for the first time, that a true heat balance must take water evaporation into account.

The most widely and universally adopted means used for cooling tower calculation is based on the theory developed in principle over 70 years ago by Merkel (1925). In this theory the sensible heat transfer because of temperature difference and the latent heat flow due to evaporation are lumped together. He used a single driving force for total heat transfer and a unique transfer coefficient. The driving force is the difference between the enthalpy of the saturated air at the interface and the enthalpy of the humid air stream. The formulation and implementation of Merkel's theory in cooling tower design and rating is presented and discussed in detail throughout most unit operations and process heat transfer textbooks. Examples are Foust et al. (1980), Coulson and Richardson (1990), Treybal (1980) and Hewitt et al. (1994). Extensive sets of curves for cooling tower design, based on Merkel's theory, have been prepared by the American Society of Heating, Refrigerating and Air Conditioning Engineers, ASHRAE (1975). However, Merkel's theory is relatively simple and based on many assumptions. The basic postulations and approximations that are inherent in Merkel's equation are (El-Dessouky et al., 1997):

(1). The resistance for heat transfer in the liquid film is negligible.

(2). The mass flow rate of water per unit of cross sectional area of the tower is constant (there is no water loss due to evaporation).

(3). The specific heat of the air-stream mixture at constant pressure is the same as that of the dry air.

(4). The Lewis number for humid air is unity.

It is important to emphasize that better analysis will lead to a better tower design and result in overall cost savings for the tower as well as trouble-free operation. Therefore, there have been many endeavors to improve upon Merkel's theory for the prediction of a cooling tower performance. The obvious source of error considered by investigators is the common practice of ignoring the effects of water evaporation. Sutherland (1983) developed a computer program to compare the accurate analysis of mechanical draught counter flow cooling towers, including water loss by evaporation. The results showed a substantial underestimate of a tower volume. The underestimation varied from 5 to 15 percent with the average value being as good as 8 percent. Nahavandi et al. (1975) showed that ignoring the evaporation losses introduces an error in the Merkel results which is not conservative and may reach up to 12 percent depending on design conditions. On the other hand, Baker (1984) cited that the effect of water evaporation is relatively small and varies with the operating conditions and gives a value for Number of Transfer Units (NTU) that is 1.34 percent lower than the actual value. The effect of water evaporation on the cooling tower performance has also been studied by Threlkeld (1970), Yadigaroglie and Pastor (1974), and Webb (1984).

Another source of errors, which has been examined, is the resistance to heat transfer in the water film and the non-unity value of the Lewis number. Jefferson (1972), Stevens et al. (1989), and Raghavan (1991) introduced an adjustment coefficient to account for the effect of the actual value of the Lewis number. Sadasivam and Balakrishnan (1995) initiated a new definition of air enthalpy, thereby avoiding the need to consider the Lewis relation. Yadigaroglie and Pastor (1974) proved that the approximation inherent in the Merkel equation contribute to the overall error. Webb

(1988) stated that none of the available analysis is totally satisfactory in (1) calculating the error of the Merkel analysis, and/or (2) defining the operating conditions within which the greatest errors exist. Accordingly, he presented a more precise method to assess the errors associated with the approximate Merkel method. He found that the approximation of unity for the Lewis number and considering the specific heat of the air-water vapor mixture as that of the dry air, result in the driving potential by a very small amount (-1.5 to 0.2 percent). He pointed out that a more complete, systematic analysis for a range of practical interest would be of value.

A simple computation method is the effectiveness-number of transfer unit (ε-NTU) approach, typically used in performance evaluation of heat exchangers. The ε-NTU method offers many advantages for the analysis of design cases in which a comparison between various types of packing must be made for the purpose of selecting the fillings best suited to accomplish a particular objective. One of the great virtues of this method is that you can develop a feel for the cooling tower performance, owing to the thermodynamic importance of the dimensionless groups in the analysis. The number of transfer units concept had been proposed earlier by Sherwood (1937) in connection with chemical engineering problems of absorption and extraction. London et al. (1940) introduced definitions of ε and NTU to use in plotting cooling tower test data. However, these definitions are not generally consistent with the basic definitions used today in heat exchanger design. They developed empirical curve fits of their ε-NTU curves for design purpose.

Moffatt (1966) is apparently the first to derive the ε-NTU equation for a counter-current cooling tower. Jaber and Webb (1989) mentioned that the Moffatt method applies only if the water is the minimum capacity rate fluid, but fails if it is not the minimum capacity rate. They developed the effectiveness-NTU design method for cooling towers. The definitions for effectiveness and NTU used by Jaber and Webb (1989) were totally consistent with the fundamental definition used in heat exchanger design. Also they presented a sample calculations for counter and cross flow cooling towers. The problem associated with the curvature of the saturated air enthalpy line was also treated. A "one-increment" design used by other investigators ignores the effect of this curvature. They demonstrated that the increased precision can be obtained by dividing the cooling range into two or more increments. The standard effectiveness-NTU method is then used for each of the increments. Calculations were presented to define the error associated with different numbers of increments. This defines the number of increments required to attain a desired degree of precision. The authors also summarized the Log-Mean Enthalpy Difference (LMED) method introduced by Berman, and showed that this is totally consistent with the effectiveness-NTU method. Berman (1961) described how the LMED method might be applied to cooling tower design. He also developed a correction factor to account for the curvature of the saturated air enthalpy curve.

El-Dessouky (1997) described a theoretical investigation for the steady state counter flow wet cooling tower with modified definition for both the number of transfer units and the tower thermal effectiveness. The modified number of transfer units is dependent on both air and water heat capacity. The effectiveness is defined by the tower cooling range and the approach. A new expression relating the tower effectiveness to the modified number of transfer units and the capacity rate ratio has been developed. The model considered the resistance to heat transfer in the water film, the non-unity of the Lewis number, and the curvature of the saturated air enthalpy curve. A procedure for implementing the model in designing or rating cooling towers has been outlined and demonstrated through illustrative examples. The model compares very satisfactorily with other methods such as LMED and conventional effectiveness-NTU. Within the range used, the obtained results showed that errors varied from +4.289 to –2.536 percent can occur in calculating the cooled water outlet temperature, and errors from +42.847 to –16.667 percent can occur in estimating the tower thermal characteristic.

Webb (1984) presented a unified theoretical treatment of the equations applicable to cooling towers, evaporative fluid coolers and evaporative condensers. This work was performed in support of ASHRAE RP-322, "Algorithms for Simulation of Cooling Towers, Evaporative Condensers and Fluid Coolers." These algorithms are described by Webb and Villacres (1984). The simulation algorithms require equations to simulate the performance of the above evaporative heat exchanger types. The key difference in the theory for each type relates to the thermal resistance of the process fluid. This thermal resistance is quite small for cooling towers, but must be accounted for in the fluid cooler and condenser. Webb (1988) showed that a "unified" theoretical treatment may be applied to all three evaporative exchanger types. Specific equations or correlations for calculation of the heat and mass transfer resistances are discussed. In the interest of preserving simplicity, the approximate Merkel equation is employed for the air side heat and mass transfer. The specific approximations associated with the Merkel equation are also noted.

It is important to emphasize that the theory presented in his work provides the basis for computer simulation algorithms for cooling towers, fluid coolers and evaporative condensers.

The effects of packing roughness on the thermal and hydraulic performance of a packed bed wet cooling tower were investigated experimentally by El-Dessouky (1996). Tower enhancement ratio and air-side pressure drop ratio were determined as functions of packing relative roughness at different values of water inlet temperature and water-to-air mass flux ratio. The packing used was glass Raschig rings 10 mm thick and 10 mm in diameter. The surface roughness was changed by adhering sand papers of different roughness to the outside surface of the packing.

The experimental data showed that an increase in the surface relative roughness from 1 to 5.62 can increase both the tower enhancement ratio and the air side pressure drop ratio up to 146% and 142%, respectively. The effect of the surface relative roughness is more pronounced at lower values of water-to-air mass flux ratio and at higher values of hot water inlet temperature. Also, a new analytical expression was developed to calculate the cooling tower characteristics.

The main objective of El-Dessouky (1996) study was to obtain quantitative information on the effect of packing surface relative roughness on cooling tower enhancement ratio and air-side pressure drop ratio. These two parameters were measured as function of hot water inlet temperature and water-to-air mass flux ratio.

Dreyer and Erens (1995) have developed a mathematical model and a computer simulation program for the modeling of counter flow, splash pack cooling tower. The one-dimensional model uses basic aerodynamic, hydrodynamic and heat-mass transfer information to predict the performance of the packing material without depending on cooling tower test data. The predicted transfer characteristics and pressure drop data obtained with the simulation program are compared with experimental data. It has been found that the model predicts the correct trends for both the transfer characteristics and the pressure drop across the packing material. Quantitatively the simulation program was found to over predict the transfer characteristics slightly. The predicted pressure drop data agreed closely to the experimental data. The program was used to obtain rough guidelines for optimum splash pack layout. The program was also employed to study the effect of reduced surface tension (resulting in smaller drops) on the thermal performance of splash pack fills.

Mohiuddin and Kant (1995) described the detailed methodology for the thermal design of wet, counter flow and cross flow types of mechanical and natural draught cooling towers. An attempt is made to present different steps of cooling tower design. The steps include selection of cooling towers; determination of tower characteristic ratio; computation of moist air properties; determination of the ratio of the water-to-air loading; integration procedure for the tower characteristic ratio. Also they have included the following design variables: the fill or packing; total height and the number of decks; water and air loading; pressure drop across the packing; natural draught tower; fan design for a mechanical draught cooling tower; blowdown and make-up water rate; water distribution system and drift eliminators. They also explained that the design of a cooling tower needs the use of different logical decisions, empirical relations and assumptions.

They have characterized the types of cooling tower in a number of different ways, such as the method of air circulation, the relative direction of air and water flow, the shape of the tower, and the heat transfer mechanism. In addition, they have also summarized eight numerical models available in the literature, and have explained in detail the integration procedure for the tower characteristic using Tchebyshev's method as suggested by the Cooling Tower Institute (CTI). Mohiuddin and Kant (1995) have classified the packing fill in modern cooling towers as two types; that is, splash type or film or cellular type. They considered two different sets of packing in their work. The first set of packing consists of eight different arrangements of decks made of redwood strips nailed to 25 mm x 51mm redwood stringers. This is essentially a splash type of packing. The second set of packing consists of 16 different arrangements of triangular splash bars, flat asbestos sheet, corrugated asbestos sheet, rectangular splash bars, asbestos louvers and typical cellular constructions.

Most cooling tower thermodynamic designs are based on an assumption of uniform water distribution across the packing. For large counter flow cooling towers, in particular natural draft cooling towers, uniform water distribution is difficult to achieve. Villiers et al. (1996) has examined the effect of non-uniform water distribution and the spatial variations of the water-to-air ratio within a cooling tower on overall thermal performance. Air temperature profiles above the fill resulting from non-uniform water distribution are presented. Theoretical versus actual results for return or re-cooled water temperatures and air temperatures above the packing are compared.

Kranc (1993) has presented a method for estimating the performance of a counter flow cooling tower packed with a regular fill. The model utilized takes into account initial non-uniformity of irrigation and the redistribution of water flow during interaction with the fill. In his analysis a local, cellular model for the heat and mass transfer has been formulated, and a computation scheme has been developed. Solutions for the non-uniform flow model have been compared to establish models that employ the assumption of uniform flow across the plan area of the tower. Results show that regions of non-uniform flow persist for some depth into the flow and that performance of the tower can be degraded.

CHAPTER 3

COOLING TOWER TYPES AND FILL

3.1 Types of Cooling Tower

Mohiuddin and Kant (1995) have summarized the classification methods for cooling towers. They showed that cooling towers can be classified in a number of different ways, such as the method of air circulation, the relative direction of air and water flow, the shape of the tower, and the method of heat transfer. It is important to mention that cooling towers are broadly classified on the basis of air circulation (i.e., the type of draft): natural draught or natural convection, mechanical draught or forced convection. On the basis of the relative movement of air and water, these towers can be grouped as cross-flow or counter-flow. According to the method of heat transfer mechanism, cooling towers are classified into three different types: wet (using evaporative cooling), dry and wet-dry.

Mechanical Draught Cooling Towers (MDCT) use fans or blowers to provide the required volume of airflow through the tower. They are subdivided into Forced Draught Cooling Towers (FDCT) and Induced Draught Cooling Towers (IDCT), depending upon the location of the fan or blower. While, the Natural Draught Cooling Towers (NDCT) use the density difference that exists between the heated, moist air inside the shell (built above the cooling tower packing) and the air outside the shell (ambient air). Typically, these towers tend to be quite large, and are frequently built in a hyperbolic contour for structural reasons. It should be noted that fan-assisted hyperbolic towers combine the features of both NDCT and MDCT.

In cross-flow towers, the air generally travels horizontally across the falling water, while in counter-flow towers it travels vertically upwards through the falling water. The counter-flow cooling towers generally provide better thermal performance compared to cross-flow type.

In wet cooling towers, cooling is achieved partly by evaporation of a fraction of the circulating water and partly by the transfer of sensible heat. While in dry cooling towers, cooling is achieved by the transfer of heat energy by convection and radiation from a hot metal surface to an air stream moving across the surface and finally rejecting the heat into the atmosphere. Wet-dry towers have the combined features of both dry and wet towers. These type of towers not only conserve water, but also minimize plume formation. In the following section we discuss the design methodology of wet cooling towers only, including both MDCT and NDCT (Hill et al., 1990).

3.1.1 Natural Draught Cooling Towers (NDCT)

Natural draught towers are rarely used today. Early designs of natural draught towers were constructed entirely of timber and were oriented to take advantage of prevailing wind; this caused obvious limitations. Introduction of the hyperbolic shape enables the chimney effect to be exploited and reduced the dependence on wind direction. The drought induced is a function of the difference in the density between the ambient air entering the bottom of the tower and the air–water vapor mixture leaving the packing. It is important to note that calculation of the operating air flow through the tower must take account of the drought induced and of the resistance to flow caused by packing and drift eliminators. The main features of a hyperbolic tower are shown in Figure 3.1.

3.1.2 Cross-Flow, Forced Draught Cooling Towers (FDCT)

Air is forced through the packing horizontally with drift eliminators on the outlet side. In these towers, axial flow fans are normally used. A simple gravity assisted hot water distribution system may be used. Modular arrangement may be made to increase capacity by mounting two or more unit side-by-side and such an arrangement facilitate control as fans can be switched on or off according to season and cooling demand. Figure 3.2 shows the main features of this tower.

3.1.3 Cross- Flow, Induced Draught Cooling Towers (IDCT)

Axial fans are normal for this arrangement also. These type of fans provide even distribution of air through the packing material compared with the forced draught design, but makes control of drift rather more difficult. Twin pack versions of this type of design

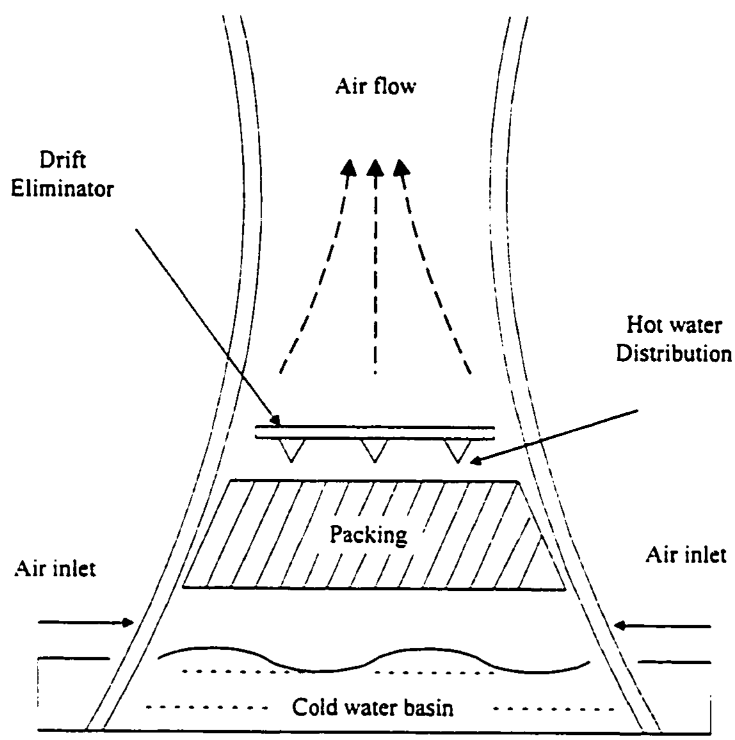

Figure 3.1: Natural draught cooling tower.

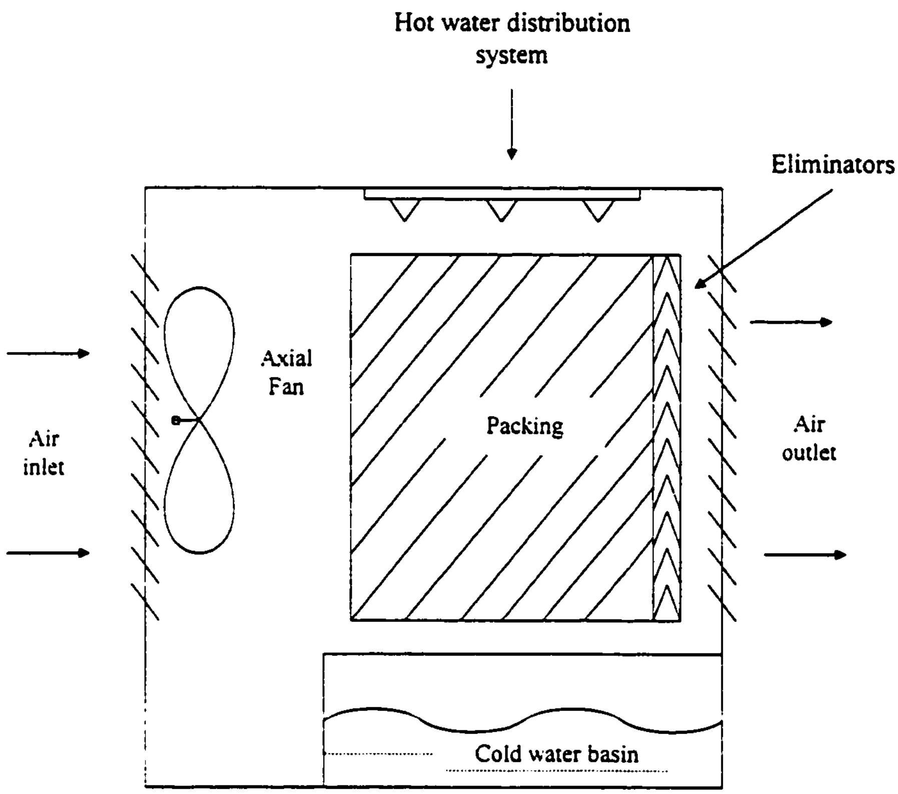

Figure 3.2: Cross-flow forced draught cooling tower.

are shown in Figure 3.3. This arrangement enables vertical discharge of the outlet air to be effective.

It should be noted that fan power for a given performance is lower as compared to forced draught design, and also a large area of drift eliminators can be accommodated. Fan motors are mounted in the warm moist air-stream and normally are sealed units.

3.1.4 Counter-Flow, Forced Draught Cooling Towers (FDCT)

Air is forced upwards through the pack by a fan mounted at low level. Axial or centrifugal fans may be used. Use of centrifugal fans enables the fan to be floor mounted with a resilient connection between fan casing and tower; such an arrangement reduces vibration and consequently noise, it also reduces the overall height of the tower. Figure 3.4 shows a schematic diagram of a counter-flow, FDCT with an axial fan.

With either fan type, re-circulation may be avoided where necessary by a canopy or directional louvers to concentrate the leaving air stream and increase its velocity. Modular designs with multiple fans may be used with fans switched on and off as needed. The use of forced draught fans facilitates indoor siting of cooling towers.

3.1.5 Counter-Flow, Induced Draught Cooling Towers (IDCT)

Axial flow fans are standard in these towers. Because the leaving air stream may be controlled in velocity and direction, re-circulation is minimized. Input air comes through louvered opening at the base of the tower and consequently performance can be affected by high winds. This may add to the airborne contaminants introduced into the cooling water. Multiple fan designs may be used, enabling one or more fans to be switched off

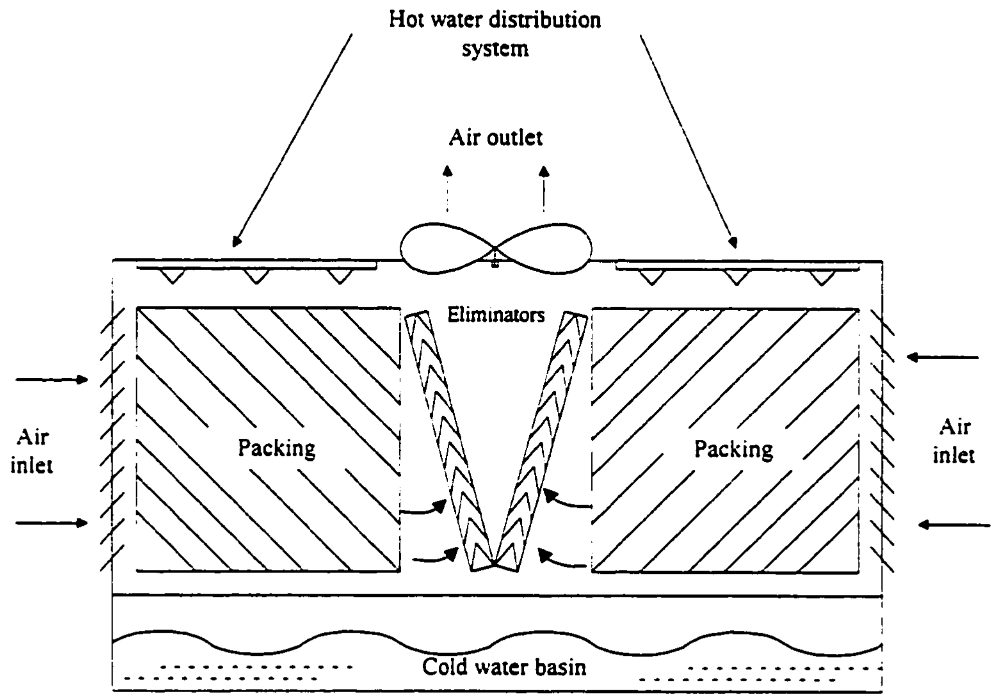

Figure 3.3: Twin packed cross-flow induced draught cooling tower.

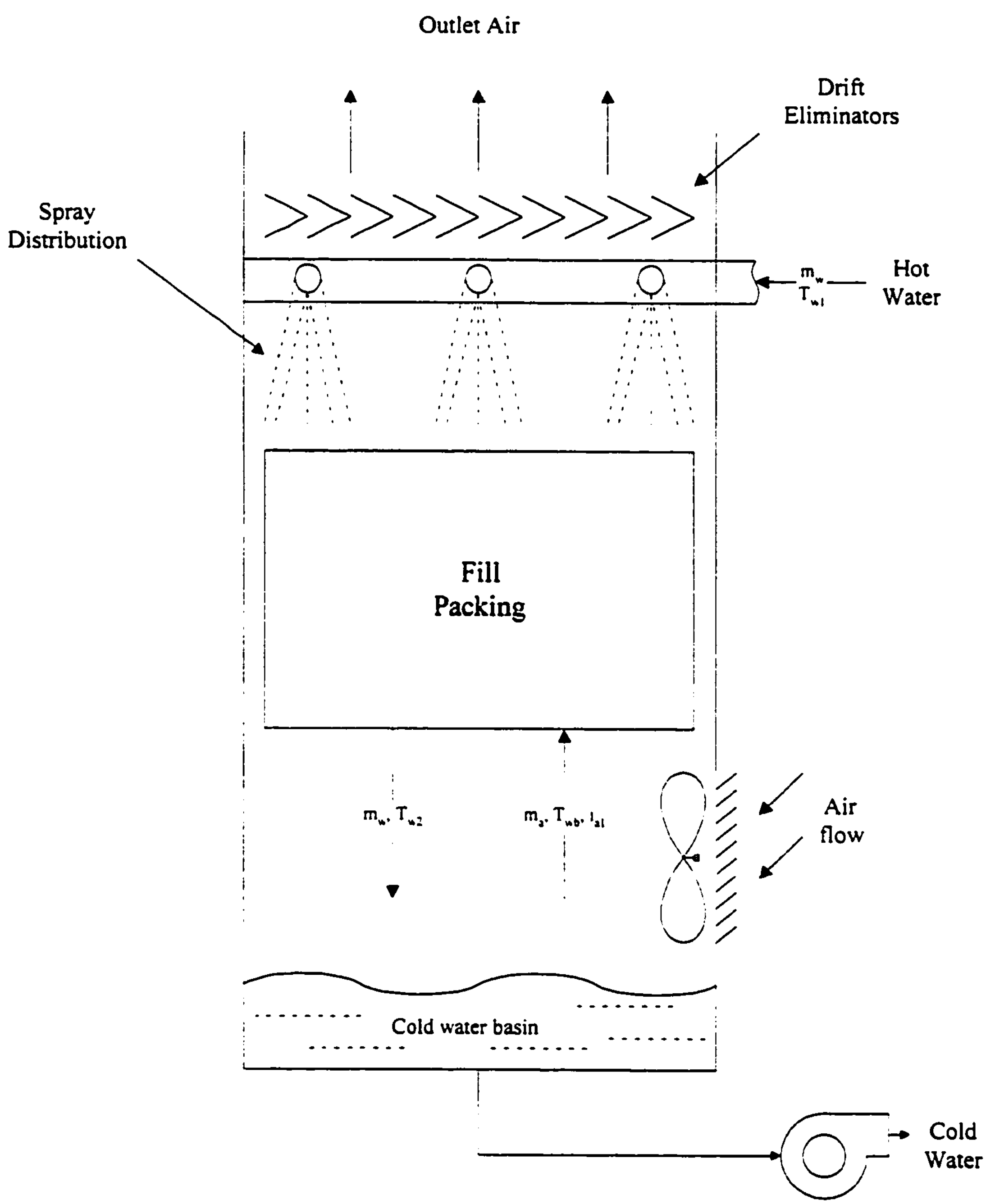

Figure 3.4: Counter-flow forced draught cooling tower.

during periods of light load. Fan motors are exposed to the warm moist air stream and must therefore be suitably protected from environment.

3.1.6 Indirect Evaporative Cooling Towers (IECT)

When applied to air conditioning systems this design incorporates a serpentine coil in the tower instead of packing. Hot water, from the refrigeration plant water cooled condenser, is circulated through the coil and cooled in the tower by the evaporative process (note that there are two independent water circuits). Although described as a closed circuit system, water is still being evaporated in the tower and cooling efficiency is lower than with packed towers; a larger tower is needed with higher capital and running costs. Contamination of the closed cooling water circuit is avoided, but purging and treatment of the tower water is still required, and is likely to be more critical. Full evaporative cooling can be achieved by placing a heat exchanger between the condenser cooling water circuit and a tower with standard packing. Figure 3.5 shows the main features of this cooling tower.

3.2 Models Available in the Literature

The physical situation within a cooling tower is very complex (films and droplets of water in air are in a constantly changing configuration). There is no mathematical model that is capable of simulating every detail of the simultaneous heat and mass transfer process occurring within the tower. Consequently, simplifying assumptions must be made for the analysis.

Eight numerical models are available in the literature: ESC code; FACTS; VERA2D; STAR; Sutherland's model; model by Fujita and Tezuka; Webb's model; and

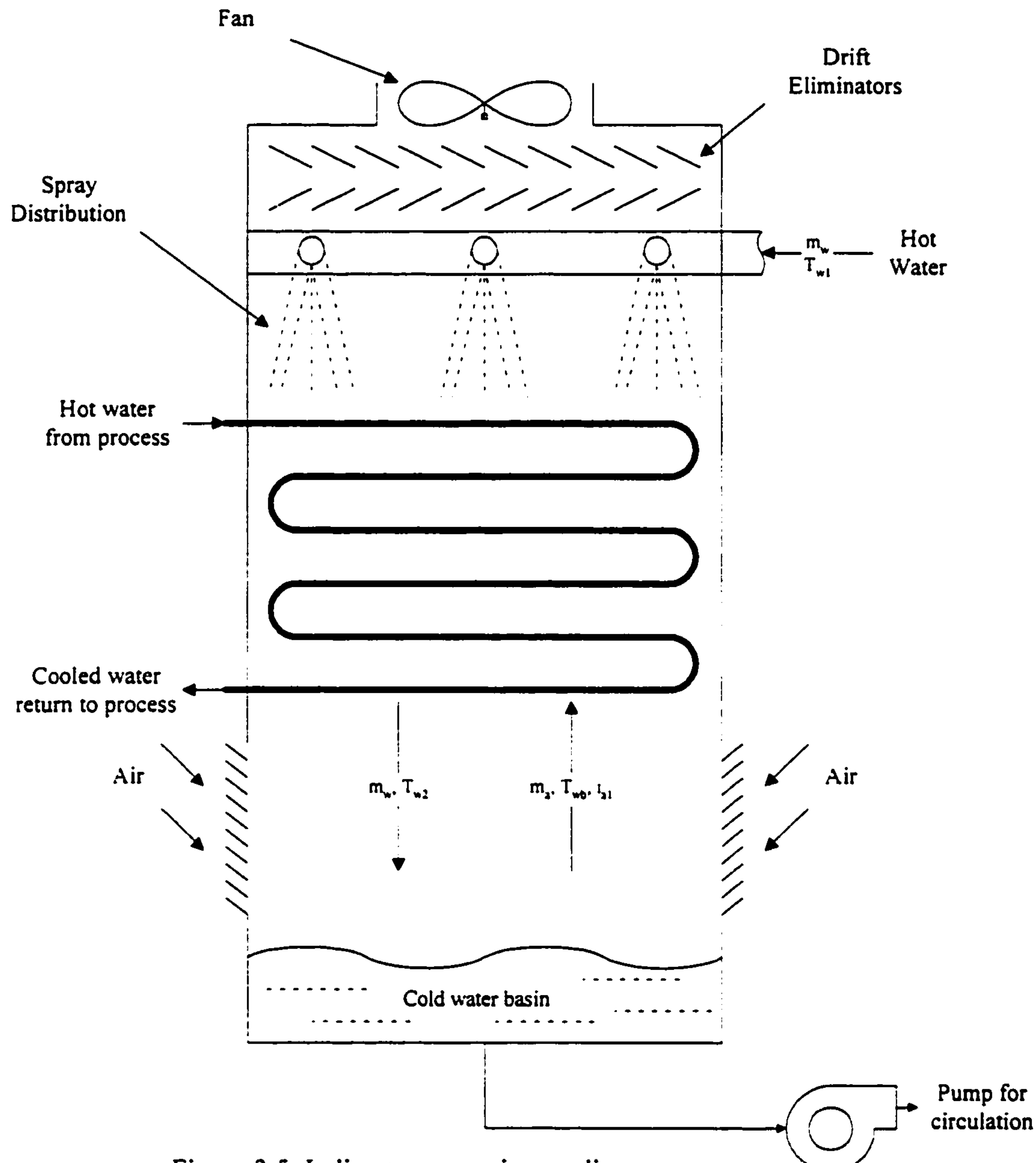

Figure 3.5: Indirect evaporative cooling tower.

model by Jaber and Webb. Each model makes use of a somewhat different set of assumptions. Consequently, the results of calculations of heat-mass transfer coefficients from each one of the models also differ.

As discussed earlier in Chapter 2, the basic theory of cooling tower operation was first proposed by Walker et al. (1923), who developed the basic equations for total mass and energy transfer, and considered each process separately. Merkel (1925) combined the coefficients of sensible heat and mass transfer into a single overall coefficient based upon the enthalpy potential as a driving force. The theory proposed by Merkel requires a few simplifying assumptions, which have been almost universally adopted for the calculation of cooling tower performance.

3.2.1 ESC code

The ESC code is based on the classical Merkel model for counter-flow and on the Zivi-Brand model (1957) for cross-flow. It is a one-dimensional model, although for cross-flow configurations it uses a two-dimensional matrix of air and water flow, but treats the flow as one-dimensional (uncoupled). Thus it is appropriate to classify this code as one-dimensional for both counter-flow and cross-flow configurations.

3.2.2 FACTS

This code is more sophisticated than a one-dimensional model, yet it contains simplifications that prevent it from being classified as a true two-dimensional code. An integral formulation of the conservation equations (conservation of mass and energy for both air and water) is applied, in conjunction with the Bernoulli equation (with head loss included). FACTS has the capability to model towers containing hybrid fill or fills that have voids or obstructions. To a limited extent, it can account for flow non-uniformities, for which FACTS offers the option of specifying a flow distribution of water at the tower inlet. It allows for the input of separate heat-mass transfer and pressure drop correlations for spray and rain regions in counter-flow towers. The FACTS code package calculates the outlet conditions of the cooling tower using the operating cooling tower parameters and packing characteristic equations.

3.2.3 VERA2D

This code treats the flow of water in the cooling tower as one-dimensional and the flow of air as two-dimensional and steady. Two-dimensional, partial differential equations are solved for the conservation of mass and energy for both air and water and the conservation of momentum for moist air. It also calculates the distribution of airflow throughout the tower. The VERA2D (Majumdar et al., 1983) code, because of the two-dimensional flow calculation, includes the following generalities: Non-uniform inlet air and water temperatures and flow-rates may be specified. The variation of air density through the tower is included as a function of temperature and pressure. Evaporation of water (which leads to non-uniform water distribution) is modeled. Heat transfer is related to both water temperature and ambient pressure. Turbulence is simulated by a local equilibrium model.

3.2.4 STAR

This code is applicable to counter-flow and cross-flow natural and mechanical draught cooling towers. It solves two-dimensional differential equations of the fluid dynamics and thermodynamics by applying a method of finite differences to a grid of rectangular mesh using a fractional step algorithm.

3.2.5 Sutherland's Model

This is a one-dimensional model developed by Sutherland (1983) for mechanical draft counter-flow cooling towers.

3.2.6 Model by Ftujita and Tezuka

This model due to Ftujita and Tezuka (1986) calculates the thermal performance of counter-flow and cross-flow MDCT using the enthalpy potential theory. The method recommends the calculation of NTU (number of transfer units) = ($K_m aV/m_w$) for counter-flow towers by the Cooling Tower Institute (CTI) method. Then the NTU for the cross-flow tower can be calculated using a correction factor.

3.2.7 Webb's Model

This model outlines an exact design procedure for cooling towers (Webb, 1988). It is a one-dimensional model, which considers water loss by evaporation. The Lewis number is different than unity.

3.2.8 Model by Jaber and Webb

Jaber and Webb (1989) described an effectiveness-NTU design method for both counter-flow and cross-flow cooling towers. They considered the LMED and ε-NTU methods to cooling tower design rather than using the standard procedure due to Merkel (1925).

The operating efficiency of each of the above codes is treated as a combination of computational efficiency and the 'user-friendliness'. Computational efficiency involves a comparison of CPU time requirements and run-time memory requirements for each of the computer model. It is difficult to draw general conclusions concerning the comparative merits of the correlations, or of the codes, yet an effort was made by Mohiuddin and Kant (1995) to compare the different models from the viewpoint of design, computational error, computational time, simplicity of usage and practicability.

3.3 Selection of a Cooling Tower

Many factors influence the selection of the type of tower (Mohiuddin and Kant, 1995). Most of them are very general at the outset of the selection procedure. A simple but effective selection process is mentioned here, which is based upon some of the major differences existing between an MDCT and an NDCT.

For very large installations, NDCTs are preferred. Nowadays, an NDCT is selected for power plants having a capacity of 500 MW and above (Mohiuddin and Kant, 1995). The initial investment on an MDCT is much lower than that of NDCT. An MDCT can be built with relatively less expensive materials than an NDCT, but the fan cost is an additional expenditure incurred in the former system. The total operation and maintenance cost favours an NDCT. In an MDCT, even though the power cost for circulating the water is less as the pumping head is less, the power cost for fans and the cost of maintaining the fans and associated equipment are considerable. Re-circulation and fogging are the major problems associated with MDCTs. Design restrictions on tower dimensions, orientation with the direction of the prevailing wind and added capacity for re-circulation can boost tower cost in the case of an MDCT. Because of the elevated discharge in an NDCT, it rarely has the problems of re-circulation and fogging. Close control of cold water temperature can be achieved in an MDCT, as the air supply in the tower can be controlled.

A closer approach and a longer range are possible in an MDCT, but in an NDCT exact control of outlet water temperature is difficult to achieve.

Choice of the flow type in a cooling tower depends on several factors (Mohiuddin and Kant, 1995). Re-circulation of outlet air, maintenance cost and cooling per unit volume of the tower are generally considered to select the proper type of flow. A cross-flow tower is thermally less efficient, owing to re-circulation of outlet air. In a cross-flow tower, an insufficient pressure head on the distribution pans also clogs the orifices because of algae and other debris that ordinarily collect in the system. So cross-flow towers have higher maintenance costs than counter-flow towers. A counter flow tower produces more cooling per unit volume for less cost than a cross-flow tower under the same design conditions. Based on the above factors, the selection procedure of the flow type in a cooling tower is considered.

When an MDCT is selected, the location of the fan determines whether the cooling tower is a Forced Draught (FD) or an Induced Draught (ID) type. Generally, cross-flow MDCTs are provided with ID fans. The draft in an MDCT is selected on the basis of the following considerations (Mohiuddin and Kant, 1995).

In an FDCT, the fan equipment is located on the ground and is easily accessible. The fans blow the air through the tower and discharge it at a relatively low velocity. In an IDCT, the fan is located on the top of the tower, which induces the flow of air through the tower and discharges it with a relatively high velocity. So, in an IDCT, because of its high discharge velocity, re-circulation is considerably reduced compared with an FDCT. Re-circulation occurs under unfavorable wind conditions when part of the hot discharge air of the tower is sucked back into the tower, which when mixed with the fresh air raises the WBT of the entering air, and thus reduces the tower performance. The high velocity discharge in an IDCT represents wasted energy in so far as the tower performance is considered. The power requirement of an IDCT is therefore greater than that of an FDCT. But the effect of re-circulation is more important in the case of close approach towers, justifying, in general, the selection of an IDCT. In the case of large approaches, it is advantageous to choose an FDCT.

Another important criterion in the selection is the level of noise intensity. If the noise intensity at ground level is a major consideration, an IDCT should be preferred. But an FDCT should be preferred if the vibration is a major consideration. In an FDCT, vibration is kept down because the mechanical equipment is installed near the ground on a solid foundation.

The maintenance cost of an IDCT fan, which is located at the top of the tower, is higher than that of a FDCT fan. If the volume of water to be cooled is large, then economics shifts in favour of an IDCT.

3.4 Cooling Tower Fill

The efficient performance of any cooling tower is dependent upon a prolonged intimate mixing or contacting of the water and air. This is accomplished by the use of fill material, which occupies a great part of the tower volume. Fill design varies with the individual manufacturers (MARLEY, 1967). However, the objective of the each manufacturer is same, i. e., to provide an efficient and economical installation. Skold (1989) discussed that the share of packing in the total cost of a tower is normally about 10 to 20%, depending on the type and size of the cooling tower. However, the use of efficient packing in a cooling tower designed for a specified thermal duty can reduce the tower diameter, height, and required floor area.

The function of cooling tower fill is to accelerate the dissipation of heat from the tower. It accomplishes this by increasing the contact time between the water and air by increasing the wetted surface and by continuously exposing the water surface to the air by promoting droplet and film formation throughout the tower.

It should be noted that tower fill material must be low in cost and easily installed. It should promote a high rate of heat transfer, should provide a low resistance to air flow, and should provide and maintain uniform water and air distribution through the life of the tower. It must be highly resistant to deterioration.

Tower filling is classified as splash type Figure 3.6 or film type Figure 3.7 (Hill et al., 1990 and MARLEY, 1967). Splash type fill is used almost exclusively in industrial cooling towers and is predominant in other tower types also. Film type fill is more adaptable to package and small commercial towers. It has the characteristic of providing more performance in less space.

3.4.1 Splash Type Fill

Splash type fill consists of many different arrangements depending on cooling tower design and manufacture. However, its purpose in any installation, is to mechanically mix the water and air. This is accomplished by the water splashing downward from one level of deck to the next with the air moving either horizontally (cross-flow) or vertically (counter-flow) through it. Maximum exposure of the water surface to the passing air is thus obtained by repeatedly arresting the water's fall, splashing it into fine droplets and spreading it into thin film on individual splash bars (Hill et al., 1990).

A prime requirement of splash type fill is that it be adequately supported. The slats or bars must remain in a horizontal position. If sagging occurs, the water and air will

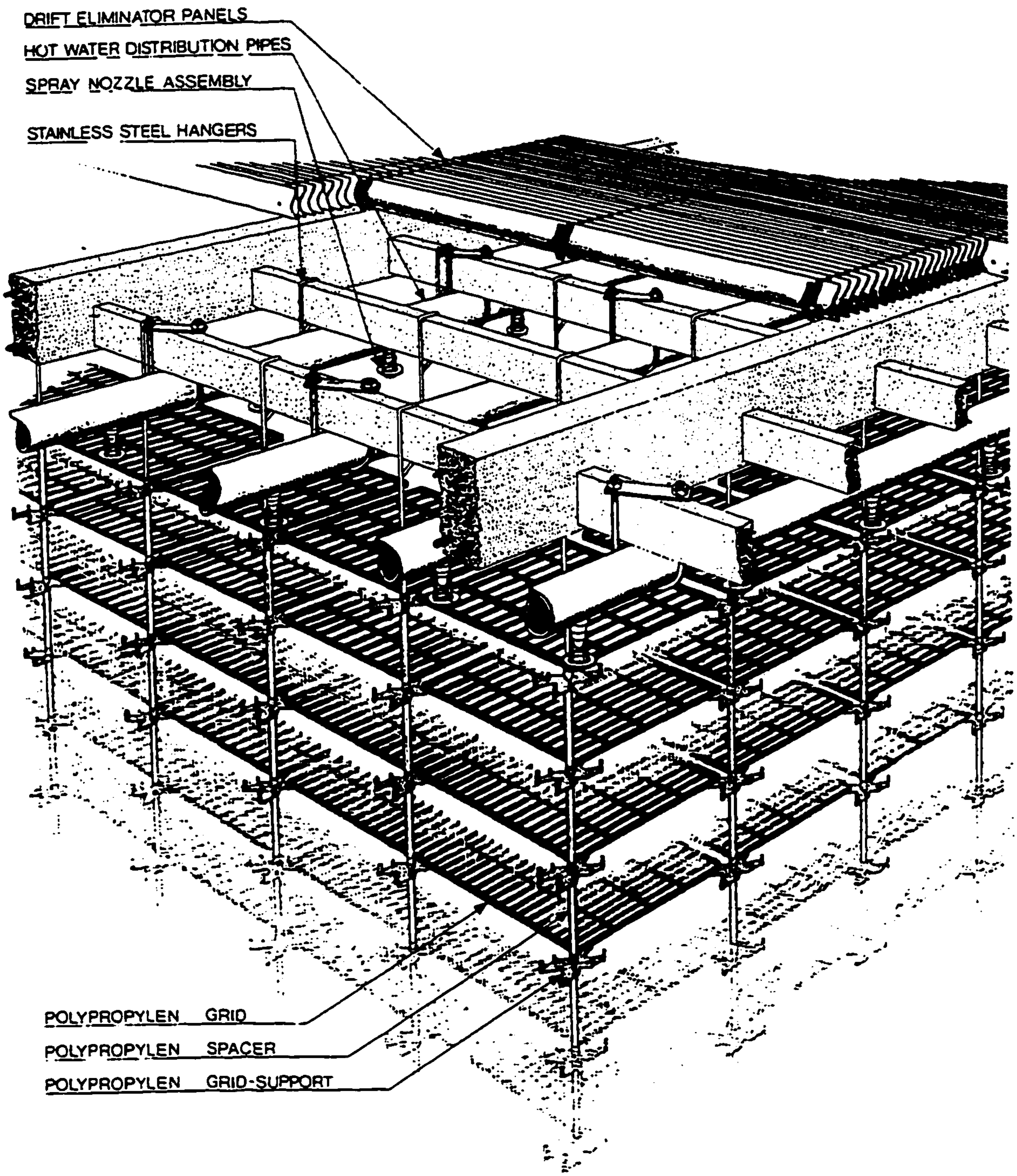

Figure 3.6: Splash type cooling tower fill. (HAMON Company)

"channel" through the tower filling and the tower performance capability will be severely decreased. This condition is most pronounced in close spaced splash type fill where the falling water more nearly approaches a drip pattern. It is equally important that the towers be level; if not, the water will tend to gravitate to the low ends of the splash bars which will also cause channeling of water and air and a decrease in tower performance (MARLEY, 1967).

Fill supports range from quality glass reinforced plastic supports spaced on close centers to simple nailed supports on wide spans. The first assures fill levelness through the life of the fill. The second is conducive to fill sagging with resulting impairment of performance.

Wood is the most popular for splash type fill application (MARLEY, 1967). It is the most economical material available for the purpose and easily installed. Clear, redwood has been popular over the years and has provided generally, satisfactory service life. Now, with the technology advancement in wood preservatives, treated woods including redwood, fir, and a few other selected species are used for this purpose and provide far longer service. Other materials, which can be used, are plastics, steel, aluminum, stainless steel, and ceramics. Because they are more expensive, their use is usually, limited to special applications

3.4.2 Film Type Fill

This type fill is gaining in use as new materials and design configurations become available (Hill et al., 1990). While more expensive than the splash type, it will provide more efficient cooling within the same amount of space. Performance of this type of fill depends upon its ability to spread the falling water into a thin film flowing over large

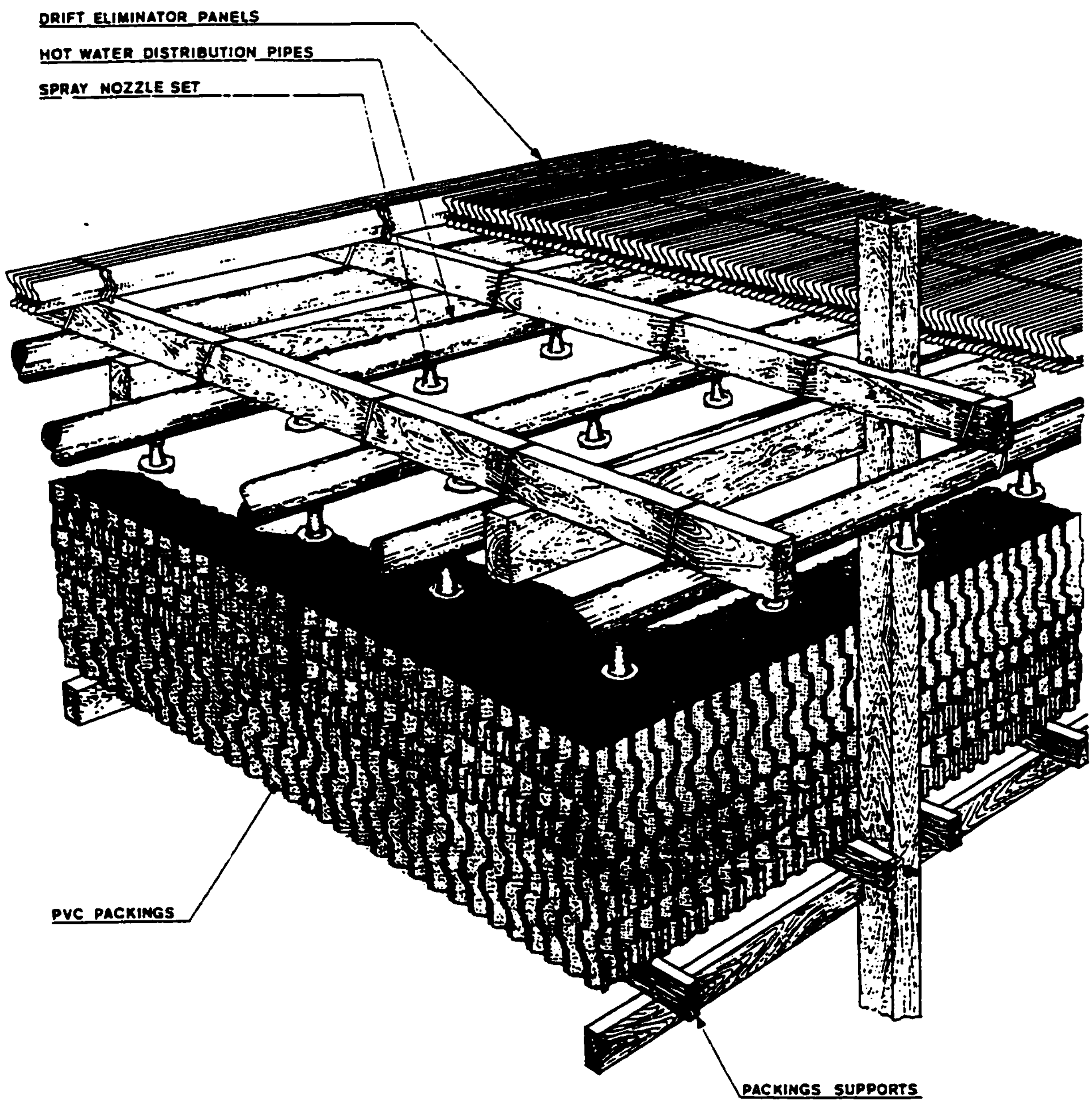

Figure 3.7: Film type cooling tower fill. (HAMON Company)

areas in order to provide maximum exposure of the water to the air stream. As it is more sensitive to irregularities of air flow and water distribution than the splash type, the tower design must assure uniform air and water flow through the whole fill area. This type of fill must also be adequately supported and uniformly spaced.

Phenolic impregnated Kraft fiber, produced in honeycomb configuration, and asbestos honeycombs are popular materials for film fill. Both are lightweight, yet structurally strong, and the latter is fireproof. Asbestos cement board in flat or corrugated sheets is sometimes used as film type fill (MARLEY, 1967). It will provide good service; however, its relatively heavy weight can present greater problems in structural design. Other materials used for film fill include plastics, aluminum, galvanized steel, and stainless steel. They are generally applied in a wave form pattern.

CHAPTER 4

MATHEMATICAL FORMULATION

4.1 Cooling Tower Theory

Figure 4.1 shows a differential control volume of a counter-flow cooling tower. In the present analysis, we have ignored water loss by drift as well as heat transfer through the walls of the tower. The total heat transfer rate between a water layer with an interface temperature of T_i and air at T_a and a humidity ratio W_a, can be expressed as (Webb, 1988):

$$dQ = K_m \{Lec_{p,a}(T_w - T_a) + i_g(W_i - W_a)\}dA \tag{4.1}$$

Considering Merkel assumption (1925), i. e. a Lewis number of unity [Le = $(h_c/c_{p,a}K_m)$ = 1] with negligible water film resistance ($T_w = T_i$), and the enthalpy of moist air as:

$$i = c_{p,a}T + i_g W$$

we can simplify equation (4.1) to give:

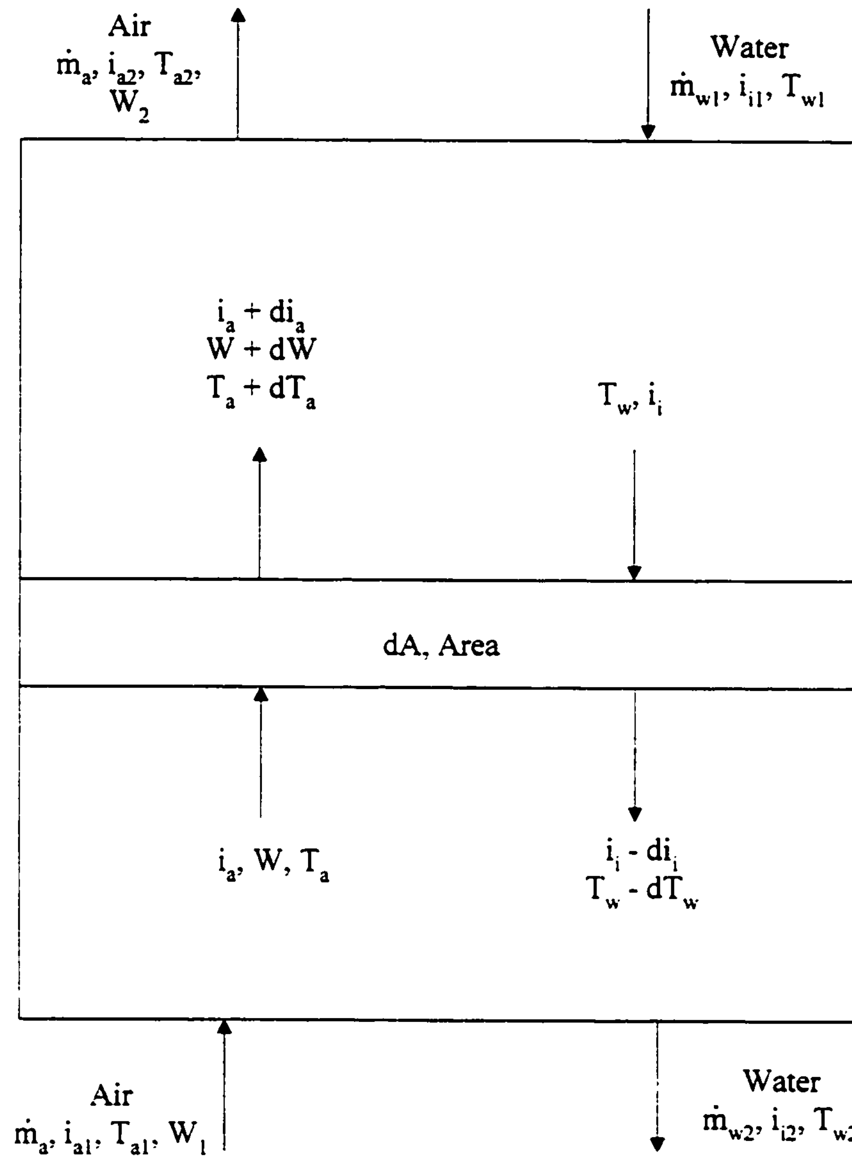

Figure 4.1: Control volume analysis of a cooling tower.

$$dQ \approx K_m(i_i - i_a)dA \tag{4.2}$$

Noting that, the heat is transferred from the water to the air stream partly as sensible heat and partly as latent heat equivalent to the fraction of water evaporated at the water-air interface. This gives,

$$dQ = \dot{m}_w c_{pw} dT_w = \dot{m}_a di_a \tag{4.3}$$

Assuming that the water interface temperature is equal to the bulk water, the above two equations were combined into a single equation. This gives, Webb (1988):

$$K_m(i_i - i_a)dA = \dot{m}_a di_a \tag{4.4}$$

Integration of equation (4.4) between the inlet and outlet positions of a cooling tower yields the following integral

$$\frac{K_m A}{\dot{m}_a} = \int_{i_{a1}}^{i_{a2}} \frac{di_a}{i_i - i_a} \tag{4.5}$$

The left hand side of equation (4.5) is called the ''Available Number of Transfer Units'' (NTU_a). It is independent of the thermodynamic conditions in the tower and is determined by the characteristic of the tower design, K_mA, and the flow rates of water and air. It is important to note that, if the transfer characteristic of a given cooling tower is known, it is possible to use the above theory to calculate the performance of the cooling tower under various air and water inlet conditions. Similarly, if the tower performance is known for specific inlet conditions (from measurements), it is possible to calculate the transfer characteristic by solution of equation (4.5).

On the other hand, the right hand side of equation (4.5) is called the ''Required Number of Transfer Units''(NTU_r). Its value depends on the operating conditions of the cooling tower, which is determined by the initial and final conditions of the air flowing through the tower. Since the relation between air saturation enthalpy and temperature is not a simple linear function of temperature, numerical integration is required to solve this integral. The Tchebycheff integration method (Cale, 1982 or Johnson, 1989) allows approximate evaluation of this relation. The effect of the assumption that the water interface temperature is equal to the bulk water temperature has been investigated by Baker and Shryock (1961), Webb (1988) and Marseille et al. (1991).

4.1.1 Calculation of the (NTU_a)

It has been stated that (K_mA/m_a) is a unique characteristic of a particular cooling tower. The NTU_a may be defined in either of the two following forms:

$$NTU_a = \frac{K_m A}{\dot{m}_a} = \frac{K_m aZ}{M_a} \tag{4.6}$$

The term K_maZ/M_a is frequently referred to as the "fill characteristic" of a cooling tower. The fill characteristic of commercially available cooling tower is a proprietary information of the manufacturers. Hence, equation or graphical representations for K_mA/m_a are not generally available. It is, however, possible to calculate the value of the 'fill characteristic' from manufacturers published cooling tower rating information. For a given cooling tower fill, the K_mA/m_a satisfies the equation:

$$NTU_a = C_1\left(\frac{M_w}{M_a}\right)^{-c_0} \tag{4.7}$$

Using the manufacturer's rating data for two different operating conditions, one can determine the coefficient (C_1) and the exponent (C_0). Once C_0 and C_1 have been determined, the value of NTU_a is easily determined from equation (4.7) for any mass flow ratio m_w/m_a. The solution of this equation generates a multiplicity of curves similar to

Figure 4.2 for different ambient and load conditions. The Cooling Tower Institute (CTI) Bulletin has over 800 of these graphs, which may be reproduced with the cooling tower algorithm presented here.

4.1.2 Calculation of the (NTU_r)

The tower characteristic is calculated using Tchebycheff's method as suggested by the CTI bulletin (1997). According to this method:

$$\int_{-1}^{1} f(x)dx = \frac{2}{n}\sum_{j=1}^{n} f(x_j) \tag{4.8}$$

where x_j indicate the real roots of the Tchebycheff quadrature polynomial for different values of n. By using this definition, one can write the NTU_r in the following form:

$$\int_{i_{a1}}^{i_{a2}} \frac{di_a}{i_i - i_a} = \frac{i_{a2} - i_{a1}}{4}\sum_{j=1}^{4}\left\{i_{i(j)} - (i_{a1} + m_{(j)}(i_{a2} - i_{a1}))\right\}^{-1} \tag{4.9}$$

where j = 1,2,3,4; $m_{(j)}$ = 0.1, 0.4, 0.6, 0.9; and $i_{i(j)}$ evaluated at $T_{w(j)} = T_{w2} + m_{(j)} (T_{w1}-T_{w2})$.

4.2 Computer Simulation

This thesis further investigates an algorithm for the performance simulation of evaporative cooling tower. This work was performed by ASHRAE, (Webb and Villacres, 1984), under the title "Cooling Towers and Evaporative Condensers". The algorithm presented is "TOWER". It is used for simulations of cooling towers (counter-flow and cross-flow types). This computer algorithm has been developed to perform rating calculations for evaporatively cooled heat exchangers.

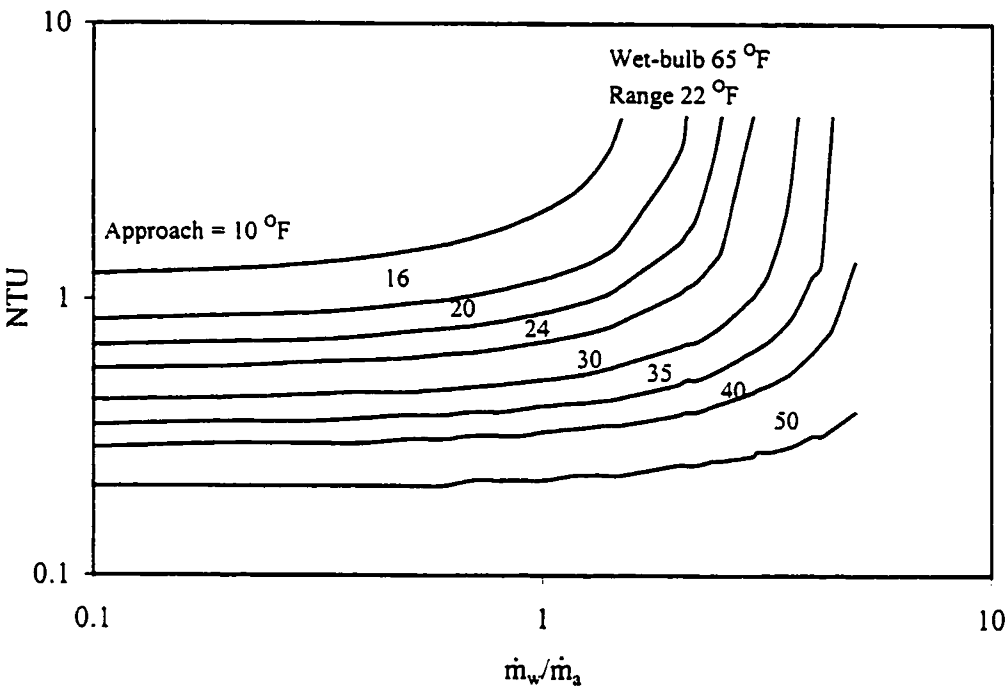

Figure 4.2: Typical characteristic curves for a counter-flow cooling tower.

The algorithm permits calculation of the cooling tower performance characteristics at off-design conditions. Although the algorithm permits computer simulation, it is not intended to replace manufacturer's rating data. The algorithm could be useful for energy systems analysis to simulate the performance of a given tower for arbitrary process and weather conditions.

The algorithm performs a "rating" calculation for a given cooling tower. The most common rating calculation seeks the heat duty of the evaporative cooling tower for specified operating conditions (fluid flow rates, inlet process conditions and ambient air wet-bulb temperature).

The first three lines of Table 4.1 define the independent operating parameters of the evaporative cooling tower considered here. The change of any of these values will influence the heat duty of a given cooling tower. Line 4 defines the heat duty in terms of the process fluid flow rate and enthalpy change. The present algorithm assumes the existence of a given evaporative cooling tower, which was designed for specific flow rates and rated at a specific process fluid inlet and air wet-bulb temperatures. Table 4.1 implies the following functional relationship between dependent and independent variables; $T_{w2} = f(T_{wb}, T_{w1}, \dot{m}_a, \dot{m}_w)$.

It is thus important to mention that, if the independent variables are specified, the dependent variables can be calculated. The results are used to determine the heat load and thus "rate" the cooling tower for the specified operating conditions. In a more general sense, one may treat any one of the variables on either side of the above functional equation as the dependent variable, and the remaining parameters as the dependent variables. For example, one may wish to calculate the required cooling tower airflow rate

Table 4.1: Operating parameter.

Parameter	Cooling Tower
1. Air conditions	Wet-bulb temperature (T_{wb}), Mass flow rate of air ($\dot{m}_a$)
2. Process conditions	Inlet water temperature (T_{w1}), Mass flow rate of fluid ($\dot{m}_w$)
3. Water flow rate	Mass flow rate of water ($\dot{m}_w$)
4. Heat duty (Q)	$C_w (T_{w1}-T_{w2})$

for specified T_{wb}, T_{w1}, T_{w2}, and $\dot{m}_w$. This shows that the rating calculation may be performed for different set of dependent variables.

The calculation scheme of the algorithm is structured to handle the major cases of possible interest. These are defined in Table 4.2. Examination of Table 4.2 shows that the definition of each case is consistent for the evaporative cooling tower, e.g., case 2 always calculates the air flow rate ($\dot{m}_a$) for specified heat duty (Q). The comment column defines specifications placed on the process fluid (that is, the water that is to be cooled).

The algorithm is designed to calculate the three "cases of interest" defined in Table 4.2. These cases are described as (Webb and Villacres, 1984):

Case 1: This is probably the case of greatest practical interest. It maintains the process fluid and spray water flow rates at the rated (or specified) values and calculates the heat duty of the heat exchanger. For a cooling tower, the leaving water temperature (T_{w2}) is the dependent variable. After calculations of T_{w2}, the algorithm calculates the heat duty from $Q = C_w (T_{w1} - T_{w2})$.

Case 2: This case calculates the airflow rate ($\dot{m}_a$) required to meet the heat load at the specified operating conditions.

Case 3: This is similar to Case 2; it calculates the water flow rate ($\dot{m}_w$) for cooling towers required to meet the heat load at the specified operating condition. One should use this case with discretion, since evaporative exchangers are not designed to permit significant variation of the process fluid flow rate.

4.3 The Algorithm for Program TOWER

As summarized in Table 4.3 and illustrated in the flow diagram, Figure 4.3 shows all three simulation cases for this program that are based on making the available number of

Table 4.2: The calculation scheme of the algorithm.

Case	Dependent Variable	Comment
1	T_{w2}	Defines Q for specified $\dot{m}_a$, T_{w1}
2	$\dot{m}_a$	Defines $\dot{m}_a$ for specified Q
3	$\dot{m}_w$	Defines $\dot{m}_w$ for specified Q

Table 4.3: Algorithm for three practical cases of cooling towers.

Case 1	Case 2	Case 3
1. Calculate $\dot{m}_w/\dot{m}_a$	1. Calculate R	1. Calculate R
2. Calculate NTU_a using Eq.4.10	2. Calculate A	2. Calculate A
3. Estimate R	3. Estimate $\dot{m}_a$	3. Estimate $\dot{m}_w$
4. Calculate A	4. Calculate NTU_a using Eq. 4.10	4. Calculate NTU_a using Eq.4.10
5. Calculate NTU_r for given tower & conditions.	5. Calculate NTU_r for given tower & conditions.	5. Calculate NTU_r for given tower & conditions.
6. Is $NTU_a = NTU_r$?	6. Is $NTU_a = NTU_r$?	6. Is $NTU_a = NTU_r$?
7. If true, print results (T_{w2})	7. If true, print results ($\dot{m}_a$)	7. If true, print results ($\dot{m}_w$)
8. If false, return to 3	8. If false, return to 3	8. If false, return to 3

Comments	Comments	Comments
The value of $\dot{m}_w/\dot{m}_a$ is fixed. The heat duty Q must be determined.	The heat duty Q is fixed. Value of $\dot{m}_w/\dot{m}_a$ must be determined.	The heat duty Q is fixed. Value of $\dot{m}_w/\dot{m}_a$ must be determined.

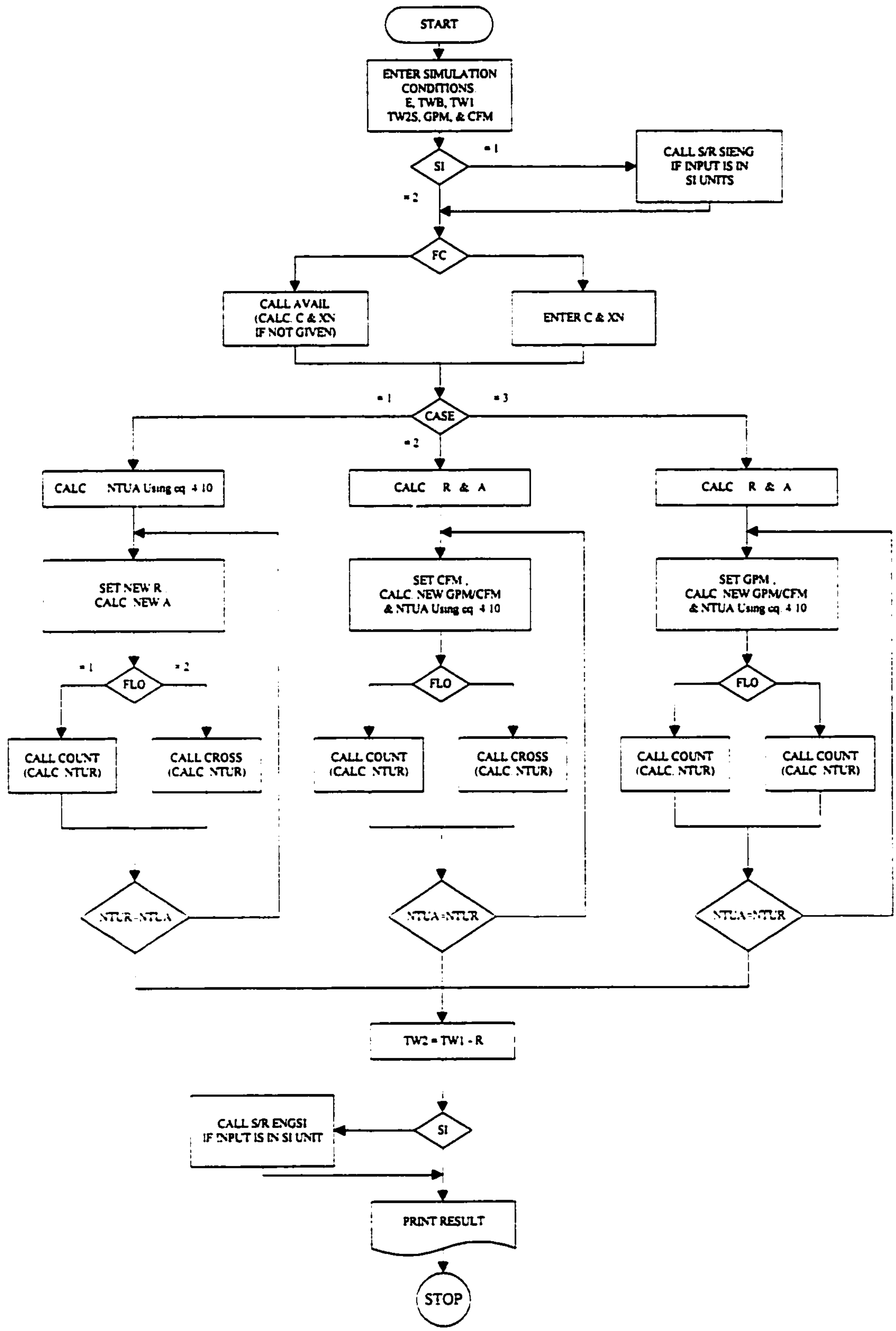

Figure 4.2: Flowchart of program TOWER (Villacres, 1984).

transfer units (NTU_a) equal the required one (NTU_r) at a certain value of $\dot{m}_w/\dot{m}_a$. This is given by:

$$NTU_a = C_1 \left(\frac{M_w}{M_a} \right)^{-C_0} \tag{4.10}$$

As can be seen from Table 4.3, the program tasks are straightforward once the coefficient C_1 and exponent C_0 of equation (4.10) are known. These characteristic values are related to the tower geometry and air velocity distribution in the fill packing. But they also depend on other factors such as the spray system, the air-water interface shape and the inlet water temperature. The last factor has been suggested by Baker and Hart (1952) and is confirmed with some of the simulations performed in this study. Regardless of what factors determine C_0 and C_1, these are proprietary values of the cooling tower manufacturers and are treated as such. ASHRAE (1979) quotes the range for C_0, $-1.1 < C_0 < -0.35$ with its average around -0.6. Because it is important for a realistic simulation to determine these values as accurately as possible, a subroutine named AVAIL has been included in the program for this purpose. The strategy used is to apply equation (4.10) to two points near the extremes of the characteristic curve of a given tower and solve the system of two equations for two unknowns (C_0, and C_1). Since the characteristic curve is not normally available, the above strategy is applied using manufacturer's catalog rating information. Once C_0 and C_1 have been determined, the value of NTU_a is easily determined with equation (4.10) for any mass flow rate ratio m_w/m_a. Thus, half of the problem has been solved.

The other half is the evaluation of NTU_r, that is, finding out the value of the integral (Mohiuddin and Kant, 1995):

$$NTU_r = \int_{i_{a1}}^{i_{a2}} \frac{di_a}{i_i - i_a} = \int_{i_{a1}}^{i_{a2}} \frac{di_a}{\Delta i} \tag{4.11.a}$$

$$NTU_r = c_w \int_{T_{w1}}^{T_{w2}} \frac{dT_w}{i_i - i_a} = c_w \int_{T_{w1}}^{T_{w2}} \frac{dT_w}{\Delta i} \tag{4.11.b}$$

This is achieved by using a different approach for the counter flow and cross flow cooling towers. For counter flow towers, subroutine COUNT uses the Tchebycheff method. According to equation (4.8) x_j indicate the real roots of the Tchebycheff quadrate polynomial for different value of n. These values can be found by the following equation (Burden and Douglas, 1985):

$$x_j = Cos\left(\frac{(2j-1)\pi}{2n}\right) \tag{4.12}$$

For $n = 4$, the roots x_j are given as ±0.187592 (≈ ±0.2) and ±0.794654 (≈ ±0.8), (Mohiuddin and Kant, 1995). Increasing the number of points in the computation of cooling tower, n, to 6 or 8 does not give much difference in the results compared with n = 4. Hence, further refinement in the number of points, n, will not bring significant effect on the computational results. Therefore, n = 4 is used in all computations. Also, it is important to mention that CTI bulletin has used n = 4 in all their computations.

According to equation (4.11), one can write:

$$T_w = \frac{T_{w1} + T_{w2}}{2} + \frac{T_{w1} - T_{w2}}{2}\delta \tag{4.13}$$

where

$\delta = -1$, $T_w = T_{w2}$, and

$\delta = +1$, $T_w = T_{w1}$

Also, $d\,T_w = ((T_{w1} - T_{w2})/2)d\delta$

Now, substituting T_w in terms of δ from equation 4.13 into equation 4.11, and inserting the corresponding limits for δ, one obtain the value of the integral I, as:

$$I = c_w \int_{T_{w1}}^{T_{w2}} \frac{dT_w}{\Delta i} = \frac{T_{w1} - T_{w2}}{2} \int_{-1}^{1} \frac{d\delta}{\Delta i} \tag{4.14}$$

Using equation (4.12), we get:

$$I = \left(\frac{T_{w1} - T_{w2}}{2}\right)\left(\frac{2}{n}\right)\sum_{j=1}^{n} \frac{1}{\Delta i_j} \tag{4.15}$$

For $n = 4$, we can write,

$$I = \left(\frac{T_{w1} - T_{w2}}{4}\right)\sum_{j=1}^{4} \frac{1}{\Delta i_j} \tag{4.16}$$

Consider the height of a counter flow tower, such that T_{w1} and T_{w2} represent the temperatures of the inlet and the outlet water (Figure 4.4). These temperatures correspond to the points +1 and −1, respectively in the interval (+1, -1) considered in the integral of equation (4.14).

In order to determine the temperature at the four specified points in the interval (+1, -1), we use equation (4.13), for the values

$\delta = -0.8$, to get $T_w = T_{w2} + 0.1(T_{w1} - T_{w2})$

$\delta = -0.2$, to get $T_w = T_{w2} + 0.4(T_{w1} - T_{w2})$

$\delta = +0.2$, to get $T_w = T_{w2} + 0.6(T_{w1} - T_{w2})$

$\delta = +0.8$, to get $T_w = T_{w2} + 0.9(T_{w1} - T_{w2})$

At these values, one can calculate the enthalpy of saturated air at the given water temperatures. Similarly, the enthalpy can be expressed as

$$i_a = \frac{i_{a1} + i_{a2}}{2} + \frac{i_{a1} - i_{a2}}{2}\delta \tag{4.17}$$

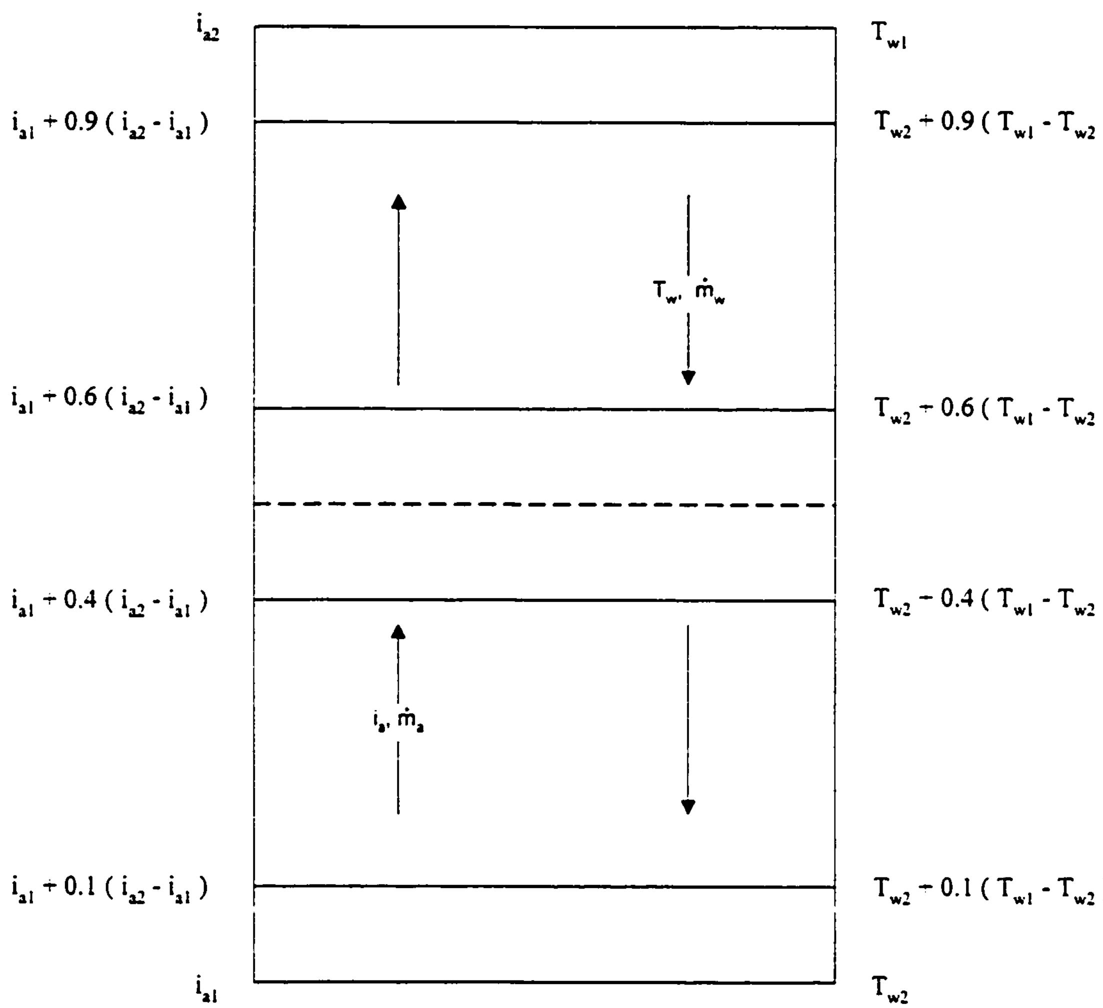

Figure 4.4: Enthalpy and temperature distribution in a counter flow tower .

Following the same procedure, the enthalpy at the corresponding sections of the tower can be obtained as:

$\delta = -0.8$, to get $i_a = i_{a1} + 0.1(i_{a2} - i_{a1})$

$\delta = -0.2$, to get $i_a = i_{a1} + 0.4(i_{a2} - i_{a1})$

$\delta = +0.2$, to get $i_a = i_{a1} + 0.6(i_{a2} - i_{a1})$

$\delta = +0.8$, to get $i_a = i_{a1} + 0.9(i_{a2} - i_{a1})$

After finding these values, one can easily calculate the value of $\sum (1/\Delta i_j)$, and hence NTU_r can be calculated.

For cross flow cooling towers, subroutines CROSS, and GRID should be used (Villacres, 1984). Subroutine CROSS divides the cross-section of the fill packing into grid of I x I squares as shown in (Figure 4.5), where I has been set as 100 but may be suitably increased or decreased, and uses auxiliary subroutine GRID to solve the energy balance equation for each individual square. The decrease in water temperature per increment is given by

$$\Delta T = 0.1\ (i_i - i_a)_{ave.} \tag{4.18}$$

This follows from the unit volume concept introduced by Baker and Hart (1952), which uses for each incremental volume, a characteristic NTU $(\dot{m}_w/\dot{m}_a) = 0.1$. has been used. According to the unit volume concept, each pair of simultaneous steps down and across increases the value of NTU $(\dot{m}_w/\dot{m}_a)$ in 0.1. Therefore, the shaded square (1,1) in (Figure 4.5.a) represents 0.1, while the four squares depicted in (Figure 4.5.d) together represent 0.2. This mechanism makes it sufficient to multiply the number of divisions of one side of the grid by 0.1 and divide by m_a/m_w to find the overall required NTU. This completes the other half of the solution.

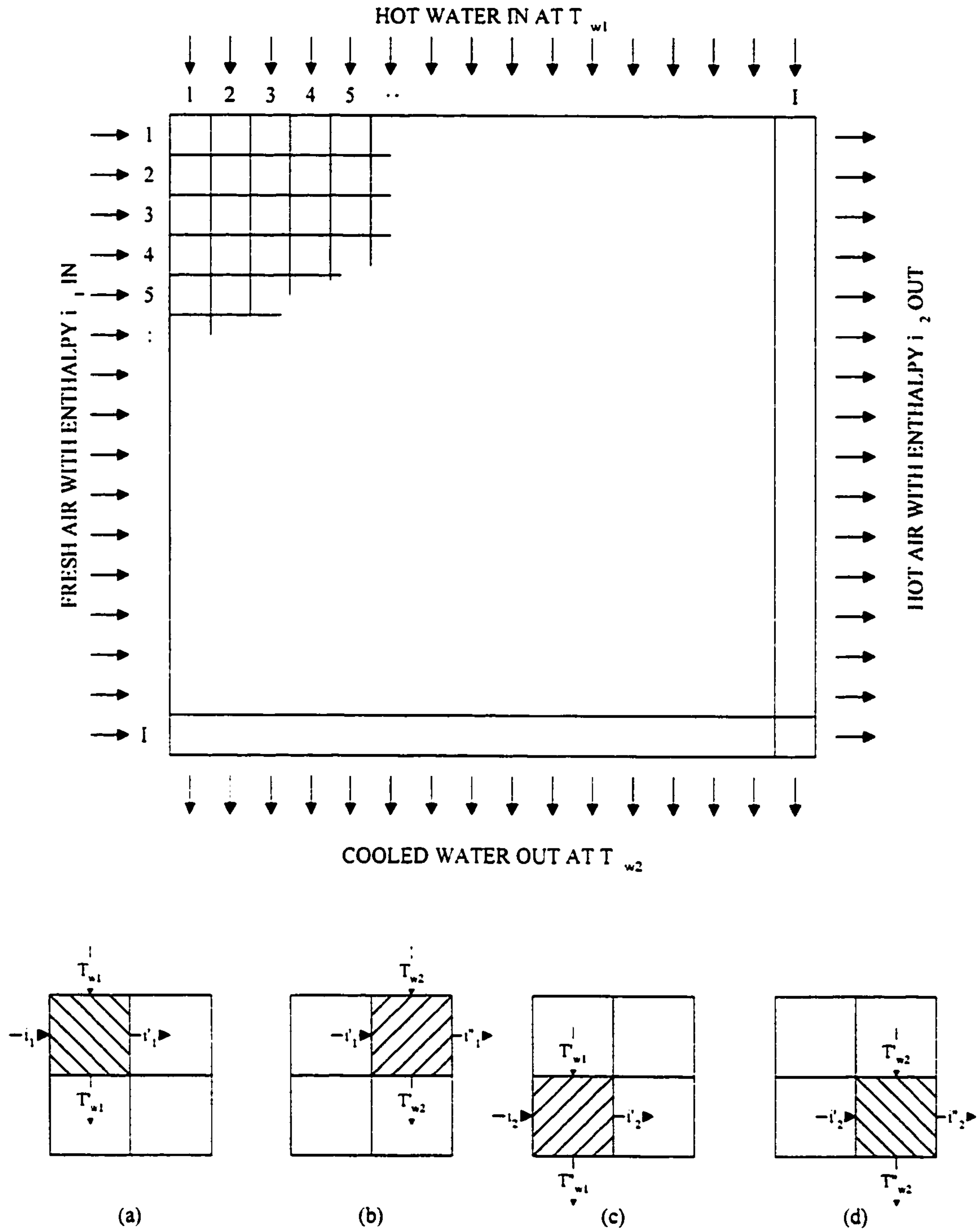

Figure 4.5: Grid used in the subroutine CROSS for the integration and the pattern used for the energy balance calculations in subroutine GRID (Villacres, 1984).

All of these calculations require the enthalpy of moist air for any combination of dry-bulb temperature and humidity ratio that may occur in the cooling tower. The difficult task of determining the moist air quality within the tower is alleviated by the use of two approximations: first, that the air enthalpy is a function of its wet-bulb temperature alone, and second, that the water-air interface is saturated air at the local bulk water temperature, meaning that $T_i = T_w$. This is also known as one of the Merkel's approximation, (Webb and Villacres, 1984).

At saturation, the wet-bulb and dry-bulb temperatures are equal. This permits the use of the following equation to calculate the saturated air enthalpy (Villacres, 1984):

$$i = T + W_s\,(2501 + 1.805\,T) \tag{4.19}$$

The humidity ratio at saturation is calculated for any altitude by

$$W_s = P_s / (P_b - P_s) \tag{4.20}$$

Equations (4.19) and (4.20), are based on perfect gas relationships and give a good approximation of the air enthalpy. They are useful for computational purpose because their use in subroutine HVTA reduces the computation time approximately threefold compared to the more exact Goff and Gratch method utilized by Kusuda (1969). The saturation pressure of water vapor is calculated with subroutine PSAT that was supplied by Walton (1983). The subroutines HVTA and PSAT have universal application to all the computer programs that are the subject of this work.

4.4 Uncertainty Analysis

Uncertainty analysis, which is frequently used in conjunction with experimentation, is adapted to the case of cooling tower performance analysis (rating and testing). Cooling towers have a number of parameters such as geometry, fluid flow rates and temperatures. The accuracy of which is to some degree uncertain. It is important to note that the main purpose of uncertainty analysis is to quantify the accuracy band for typical performance parameters as a result of uncertainties associated with input variables and to identify the most important contributors to the overall uncertainty (Irfan, 1997).

4.4.1 Formulation

In general any independent variable X can be represented as (Irfan, 1997):

$$X = X \pm U_x \tag{4.21}$$

where X denotes its nominal value and U_x its uncertainty about the nominal value. The $\pm U_x$ interval is defined as the band within which the true value of the variable X can be expected to lie with a certain level of confidence.

Basically uncertainty in an independent parameter X is composed of two components, precision error (P_x) and bias error (B_x). Precision error represents an estimate of the lack of repeatability caused primarily due to randomness or unsteadiness whereas bias error gives an estimate of the fixed or constant error. The uncertainty U_x is then given in terms of these components as (Irfan 1997):

$$U_X = \sqrt{P_X^2 + B_X^2} \tag{4.22}$$

It should be noted that for calibrated instruments the extent of bias error is negligibly small and is therefore normally neglected.

In general, if a function Y(X) represents a response variable, then the uncertainty in Y due to an uncertainty in X is expressed in a differential form as

$$U_Y = \frac{dY}{dX} U_X \tag{4.23}$$

For a multivariable function $Y = Y(X_1, X_2, X_3 \ldots, X_N)$, the uncertainty in Y due to uncertainties in the independent variables is given by the root sum square product of the individual uncertainties computed to first order accuracy as (James et al., 1995):

$$U_Y = \left[\sum_{i=1}^{N} \left(\frac{\partial Y}{\partial X_i} U_{X_i} \right)^2 \right]^{1/2} \tag{4.24}$$

Physically, each partial derivative in the above equation represents the sensitivity of the parameter Y to small changes in the independent variable X_i. The partial derivatives are therefore also referred to as *sensitivity coefficients.*

By normalizing the uncertainties in the resultant parameter Y and the various input variables by their respective nominal values, equation (4.24) can be written as

$$\left(\frac{U_Y}{\overline{Y}} \right) = \left\{ \sum_{i=1}^{N} \left[\left(\frac{\partial Y}{\overline{Y}} \frac{\overline{X}_i}{\partial X_i} \right) \left(\frac{U_{X_i}}{\overline{X}_i} \right) \right]^2 \right\}^{1/2} \tag{4.25}$$

The dimensionless terms in braces on the right hand side of the above equation represent the respective sensitivity coefficients and uncertainties in their normalized forms and are therefore referred to as normalized sensitivity coefficients and normalized uncertainties denoted by NSC and NU, respectively (James, 1995). Equation 4.25 can therefore be written as

$$\left(\frac{U_Y}{\overline{Y}} \right) = \left\{ \sum_{i=1}^{N} \left[\left(NSC_{X_i} \right) \left(NU_{X_i} \right) \right]^2 \right\}^{1/2} \tag{4.26}$$

A dimensionless factor ε_{xi} is introduced to represent the positive and negative uncertainty in the variable X_i such that

$$U_{X_i} = \overline{X}_i \varepsilon_{Xi} \tag{4.27}$$

With the help of this substitution and on replacing partial derivatives by ratios of discrete changes, the normalized sensitivity coefficients and uncertainties can be expressed as

$$NSC_{X_i} = \frac{\Delta Y_i}{\overline{Y}} \frac{\overline{X}_i}{\Delta X_i} \tag{4.28a}$$

$$NU_{X_i} = \varepsilon_{X_i} \tag{4.28b}$$

ΔX_i in the above equation can be written as

$$\Delta X_i = \overline{X}_i(1 + \varepsilon_{X_i}) - \overline{X}_i(1 - \varepsilon_{X_i}) = \overline{X}_i(2\varepsilon_{X_i})$$

Therefore, equation (4.26) now becomes

$$\varepsilon_Y = \left\{ \sum_{i=1}^{N} \left[\left(\frac{\Delta Y_i}{\overline{Y}} \frac{1}{2\varepsilon_{X_i}} \right) \left(\varepsilon_{X_i} \right) \right]^2 \right\}^{1/2} \tag{4.29}$$

Another parameter of interest is the relative contribution of each input variable uncertainty to the overall uncertainty in the dependent variable defined by James et al. (1995) as:

$$\text{Relative Contribution} = RC = \left(\frac{(NSC_{X_i})(NU_{X_i})}{\varepsilon_Y} \right)^2 \tag{4.30}$$

An examination of above equations shows that the propagation of the uncertainty in a particular input parameter through the analysis equations into the result is dependent on the magnitude of the normalized sensitivity coefficients. If the NSC of a variable is of the order of unity then its uncertainty, on being squared, is propagated essentially unchanged. If it is greater than unity its uncertainty is amplified whereas if it is less than unity, its effect is diminished. Moreover, since the sensitivity coefficients of the various input variables are normalized relative to the same nominal value Y, a one on one comparison of the coefficients can be made thereby yielding a good estimate of the sensitivity of the result to each of the variables.

Relative contribution of a variable to the overall uncertainty involves the square of the product of its normalized sensitivity coefficient and uncertainty. Consequently, it is their product that is of significance and not the individual terms themselves. Relative contribution of any variable to the overall uncertainty can be controlled to a large extent by bringing down the uncertainty of that variable. It is thus important to note that the normalized sensitivity coefficients and relative contributions are obtained as significant characteristic parameters in the uncertainty analysis of any dependent variable. While the sensitivity coefficients identify the input parameters to which the performance parameters are most sensitive, irrespective of the uncertainty in the input variables themselves, the relative contributions identify the dominant uncertainty contributors.

4.4.2 Computation Scheme

The steps required for implementing the uncertainty analysis scheme are outlined here. First the various input variables to be examined are identified. The nominal value of the performance parameter Y is then determined by using nominal values of all the input variables. Then each input variable is perturbed on either side of its nominal value and the corresponding values of the output parameter for each perturbation are recorded. The difference in the two values of Y gives the change ΔY_i for that input variable and can be expressed as

$$\Delta Y_i = Y(X_1, X_2, \ldots X_I(1+\varepsilon_{xi}) \ldots, X_N) - Y(X_1, X_2, \ldots X_I(1-\varepsilon_{xi}) \ldots, X_N) \qquad (4.31)$$

where N is the number of input variables considered. The uncertainty in Y is then determined by using equation (4.29). It should be noted that when a particular variable is subjected to perturbation, the values of the rest of the input variables are held at their respective nominal values, as shown schematically in Figure 4.6.

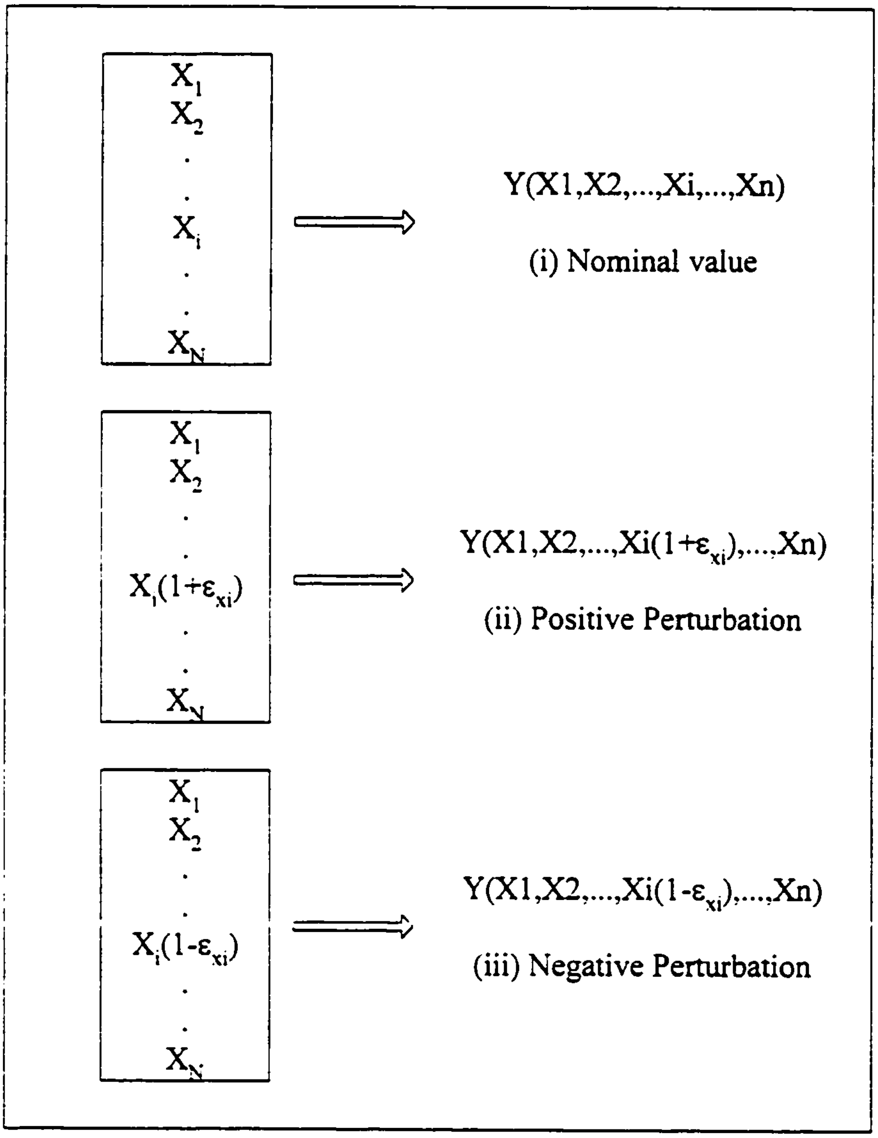

Figure 4.6: Schematic diagram showing the two-way perturbation of input variables about their nominal values (Irfan, 1997).

CHAPTER 5

RESULTS AND DISCUSSIONS

The computational result presented in this section are intended to predict the effect of different cooling tower fill packing types, fluid flow rates and temperatures on the cooling tower performance characteristics. The performance results are also compared with the experimental data, in addition to the sensitivity and uncertainty analysis of the important parameters such as wet-bulb temperature, inlet water temperature, and mass flow rates of air and water..

5.1 Sources of Experimental Data

The experimental data to be presented were obtained from Simpson and Sherwood (1946) of a small cooling tower that was used in connection with the installation of absorption-type air conditioning units for home and small stores. The design was restricted by the expected load, the limited headroom for installation in the basement, etc., the requirement of low or moderate pressure drop across the blower and by the inlet air wet-bulb temperature as high as 85 ^{O}F.

Simpson and Sherwood (1946) data are reported on three tower designs, designated as R, M, and C. The dimensions of the towers are summarized in Table 5.1. The principle difference between the several towers was the nature of the internal packing over which the water was distributed. Narrow redwood slats was used in R tower; parallel vertical Masonite sheets in M tower; and 2-inch ceramic rings in C tower.

In the case of R tower, the slats were arranged in parallel and one directly above another with free vertical air passages, 3/8 inch wide, between vertical slat assemblies. In the case of M tower, the Masonite sheets were set parallel and vertical. The sprays were 4.625 inch above the top of the packing and the water level in the pan was about 9 inch below the bottom of the packing. In the case of C tower, it was packed with Rasching rings, these rings are hollow cylinders with a height equal to their outer diameter.

In addition to the types of tower mentioned above, a Baltimore Air Coil model VXT-470 of the counter-flow type is simulated by the algorithm discussed earlier in Chapter 4. The results are compared with the catalog rating data given by Villacres (1984). Also a recent data has been taken from CTI bulletin (1997). This data represent an actual cooling tower working in Houston.

Table 5.1: Physical data on experimental cooling tower and packing materials.

	Redwood slats (R)	Masonite sheets (M)	Ceramic rings (C)
Tower width (in.)	41.625	41.625	41.625
Tower depth (in.)	23.75	23.75	23.75
Tower height (in.)	84	84	84
Packed height (in.)	41.375	41.375	41.375
Packed material	Redwood slats	Masonite sheets	Ceramic rings

5.2 *Validation of the Calculation Procedure*

The parameter that affect the performance of cooling tower are (Hill et al. 1990):

- The wet-bulb temperature
- The inlet water temperature
- The flow rate of water
- The flow rate of air

To validate the numerical procedure, the computational results for the outlet water temperature were compared with experimental results available in the literature by varying the parameter mentioned above. The comparison here covers the R, M, and C types of tower, and Baltimore Air Coil model VXT-470.

The cooling tower, Baltimore Air Coil model VXT-470 of the counter-flow type, is simulated as shown in Figure 5.1 for the rated air and water flow rates. Figure 5.1, also shown by Villacres (1984), shows the comparison of the predicted and catalog rating values of the outlet water temperature when the inlet water temperature is changed. As can be seen, the outlet water temperature increases with the increase in inlet water temperature. The agreement is good with that the maximum percentage difference is 0.86. This occurs at the hot water end because the temperature differential between the water film and the interface increase with higher hot water temperatures.

The tower packed with the Redwood slats of the counter flow type is also simulated by the program devolped in this work. The computational results are compared

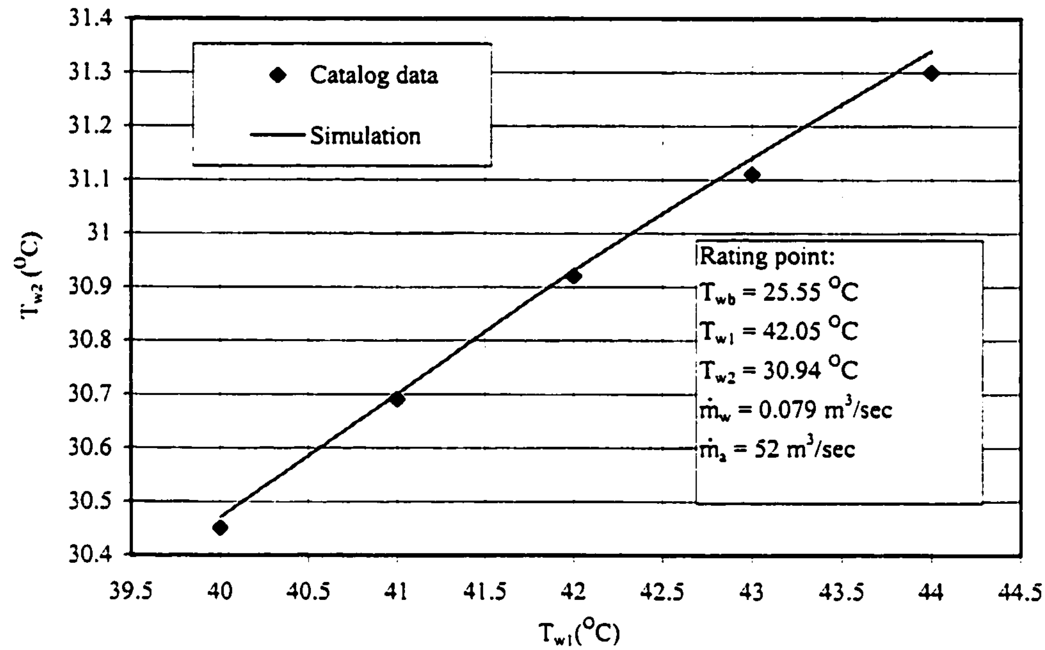

Figure 5.1: Effect of the inlet water temperature on the outlet water temperature in the BAC VXT-470 of a counter-flow tower.

with Simpson and Sherwood (1946) results for the rated air and water flow rates. As can be seen from Figure 5.2 the predicted results are in good agreement with the experimental results. The maximum difference in the predicted result is 1.2%.

The same tower which was used by Sherwood and Simpson (1946) is also simulated but with different fill packed material. Figure 5.3 shows the comparison of the predicted and the experimental results of the tower packed with ceramic rings for the rated wet-bulb temperature and mass flow rate of water. As can be seen, the outlet water temperature decreases with the increase in the mass flow rate of air. Because the rate of heat transfer gets enhanced by increasing the mass flow rate of air. Close agreement is exhibited between the predicted and experimental results.

Performance curves are then generated and compared with the catalog data (Villacres, 1984) for the Baltimore Air Coil tower for different wet-bulb temperatures. These results are shown in Figure 5.4. It can be seen from the figure that the wet-bulb temperature has the same effect as the inlet water temperature on the outlet water temperature. We found that the computed values are in good agreement with the experimental results. Also, Figure 5.5 shows a good agreement, in trend, as well as the absolute values when the simulated results are compared with the experimental results for a cooling tower packed with redwood slats.

In order to compare the effect of mass flow rate of water on the outlet water temperature, the results available in the literature is for a cooling tower packed with masonite sheet. Figure 5.6 shows the comparison between the calculated and experimental

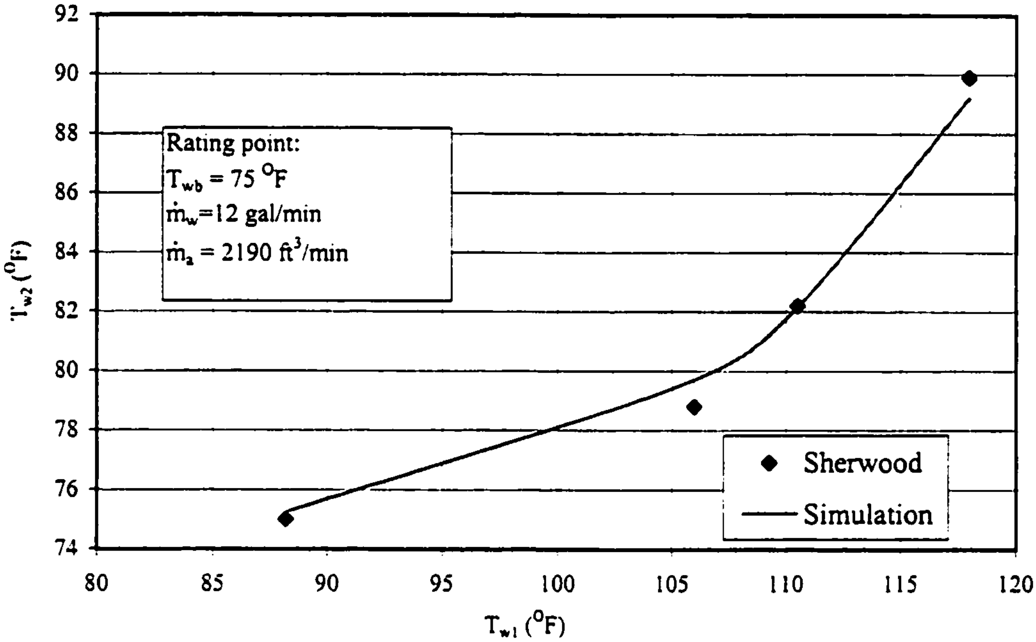

Figure 5.2: Effect of the inlet water temperature on the outlet water temperature in the tower packed with Redwood slats.

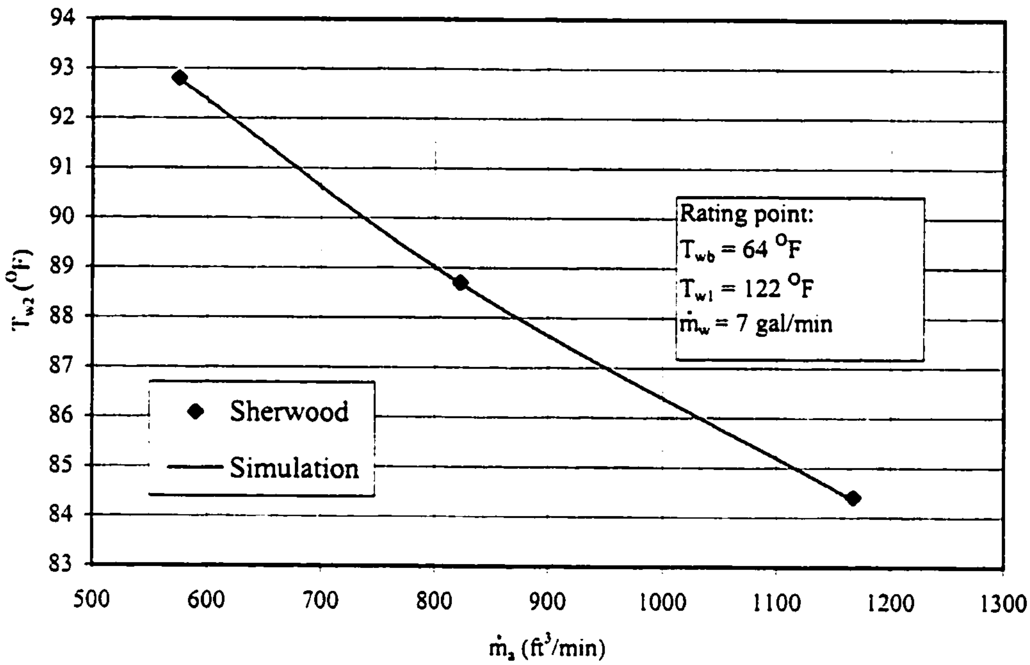

Figure 5.3: Effect of the mass flow rate of air on the outlet water temperature in the tower packed with Ceramic rings.

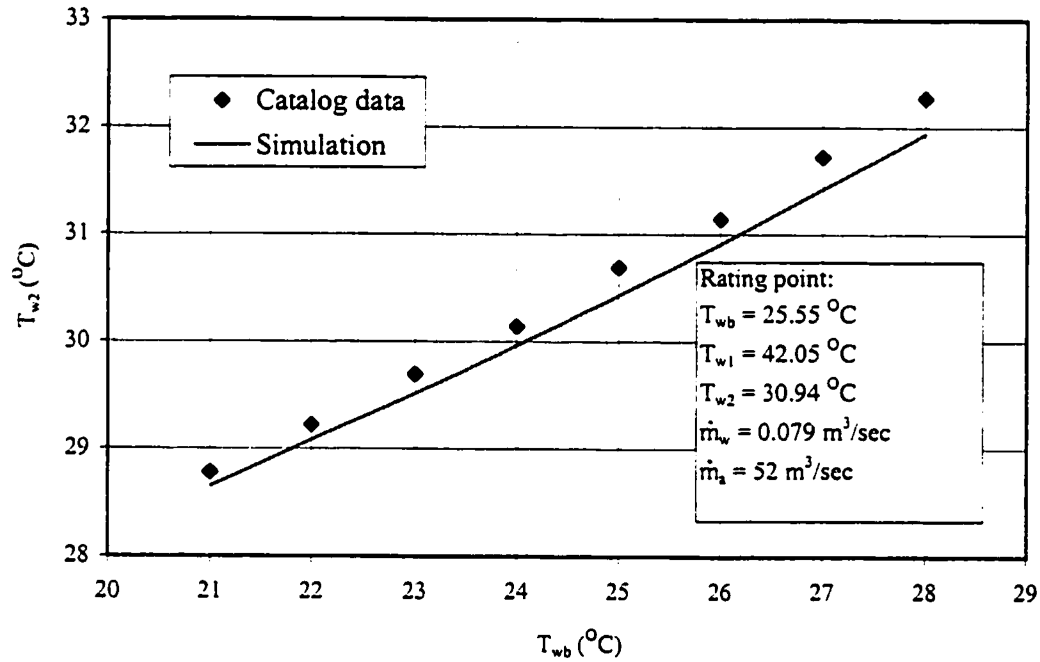

Figure 5.4: Effect of the wet-bulbtemperature on the outlet water temperature of BAC VXT-470 counter-flow tower.

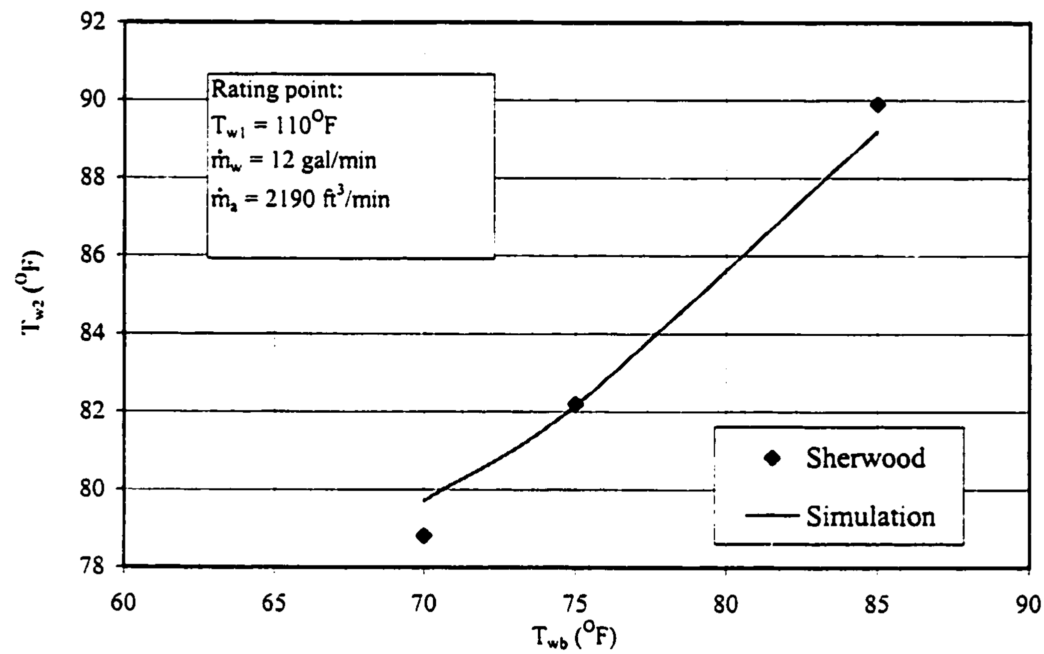

Figure 5.5: Effect of the wet-bulb temperature on the outlet water temperature in the tower packed with Redwood slats.

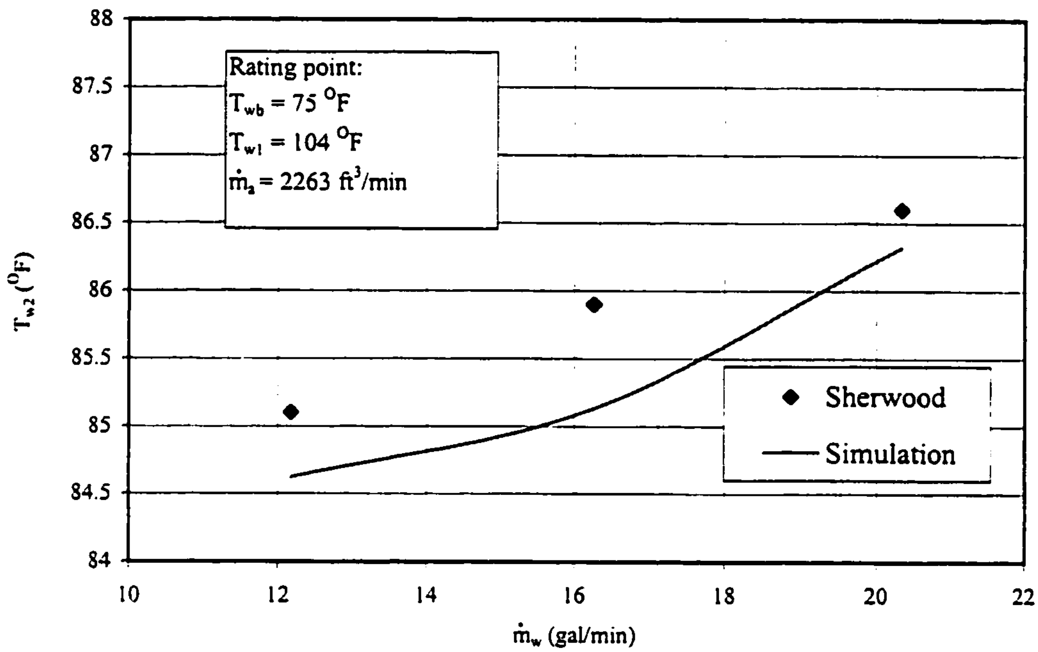

Figure 5.6: Effect of the mass flow rate of water on the outlet water temperature in the tower packed with Masonite sheet.

values for the rated inlet water temperature, wet-bulb temperature, and for different mass flow rates of air.

5.3 Results of the Cooling Performance

In this section, an investigation is carried out on different types of fill packing material available in the literature as well as from different manufacturers. Also, a comparison is made for the thermal characteristics of a cooling tower when a different type of fill packing system is used. In addition to the above investigation on fill packing material, a sensitivity analysis on the thermal characteristics of cooling tower is also studied with respect to the important parameters such as inlet water temperature, wet bulb temperature, and mass flow rates of air and water.

5.3.1 Effect of different parameters on cooling performance

As discussed earlier, Sherwood and Simpson (1946) have done an experimental work on a small draught cooling tower packed with different fill materials. The experimental data is used to study the effect of each parameter on the cooling performance while keeping the other parameters constant. A comparison is made for the three different fill packed system discussed in the literature.

The inlet water temperature is varied from 85 to 125 ^{O}F, the wet bulb temperature ranged between 60 and 85 ^{O}F, the mass flow rates of air and water varied as $1100 < \dot{m}_a < 2600$ ft^3/min and $9 < \dot{m}_w < 20$ gal/min. The range, which is used in the simulation for each parameter, is defined as the difference between the extreme values discussed earlier in the work of Sherwood and Simpson (1946).

It is important to note that the inlet hot water temperature and wet bulb temperature have a direct relationship with the outlet water temperature, which can be seen in Figures 5.7 and 5.8, respectively. Figure 5.9 shows the effect of the mass flow rate of water on the outlet water temperature. For all the fill packed materials studied, as we increase the mass flow rate of water the outlet water temperature increases. Figure 5.10 shows the effect of the mass flow rate of air on the outlet water temperature for the three different fill packed materials. As can be seen, the mass flow rate of air is the only parameter that gives an inverse relationship with the outlet water temperature.

For all the parameters studied, it can be concluded that the Redwood slat is the most efficient material because it gives the lowest outlet water temperature. The parametric study reveals that lower the inlet water temperature and wet-bulb temperature better would be the performance of cooling tower. Another way to increase the performance is to increase the mass flow rate of air.

The simulation is then extended to evaluate the performance of an actual cooling tower. The data for this tower is taken from CTI bulletin (1997). The tower is a counter flow splash fill type. The capacity of the tower is 10,000 gal/min, and the mass flow rate of water to air ratio is 1.37. The design inlet and outlet water temperature is 110 and 90 ^{O}F, respectively, with 80 ^{O}F wet bulb-temperature.

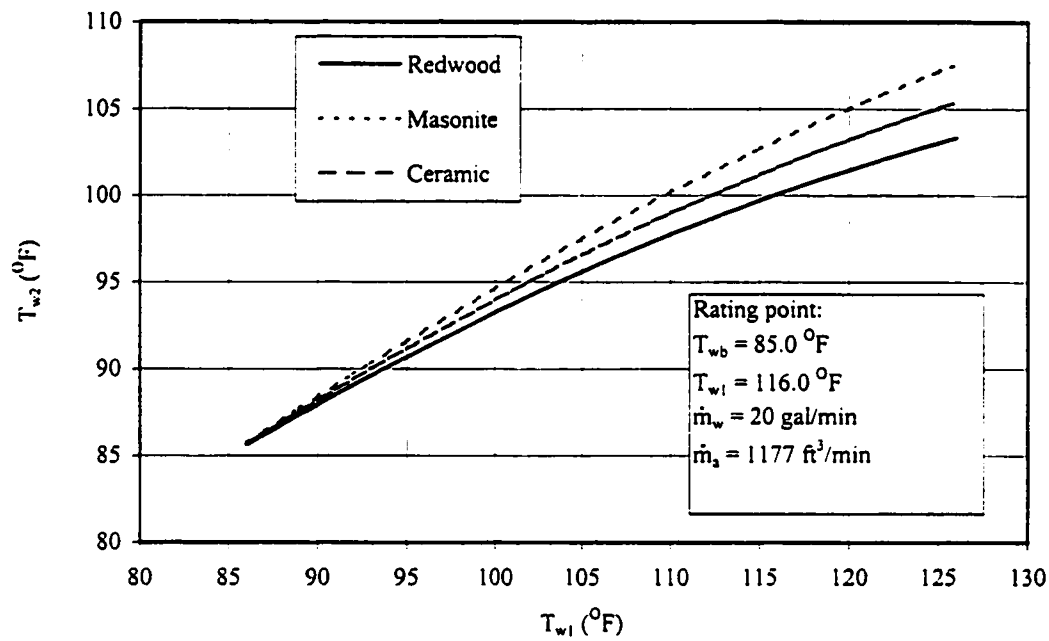

Figure 5.7: Effect of the inlet water temperature on the outlet water temperature for the three different fill packed systems.

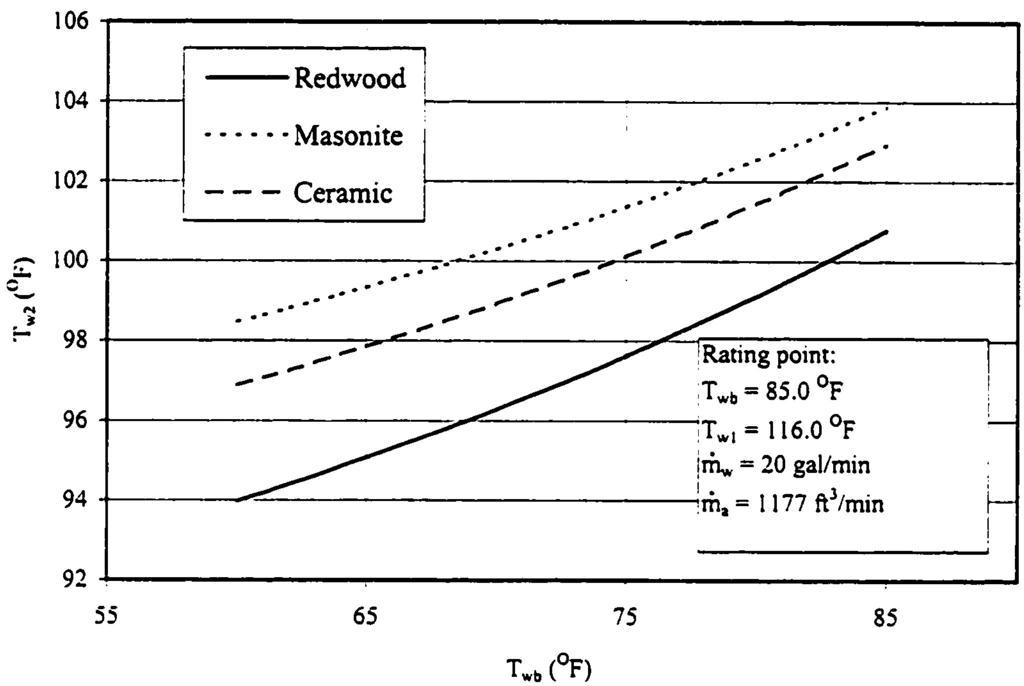

Figure 5.8: Effect of the wet-bulb temperature on the outlet water temperature for the three different fill packed systems.

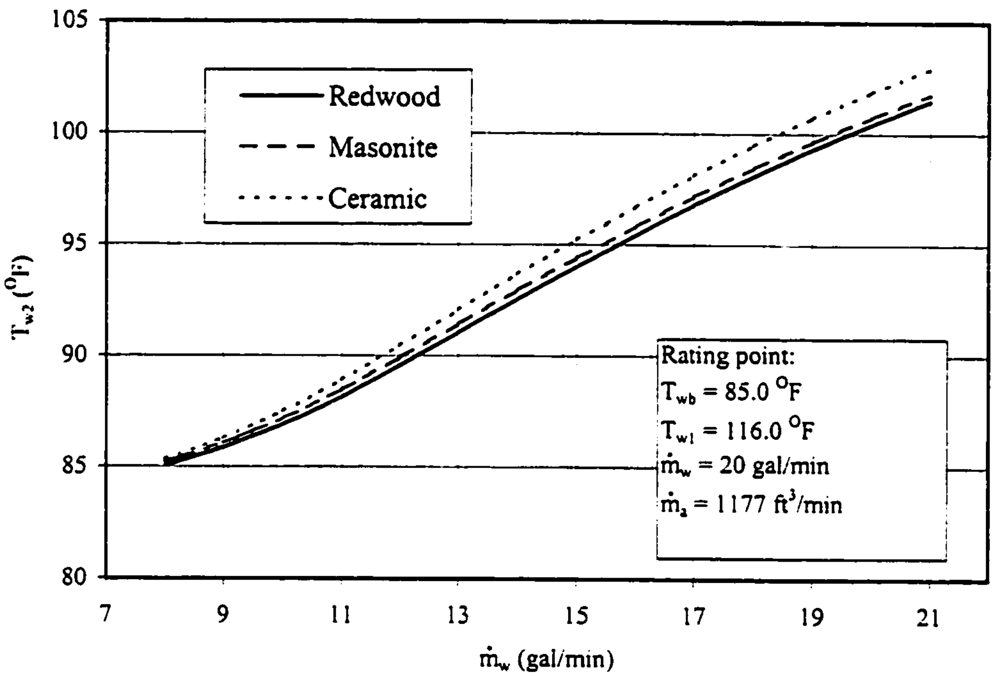

Figure 5.9: Effect of the mass flow rate of water on the outlet water temperature for the three different fill packed systems.

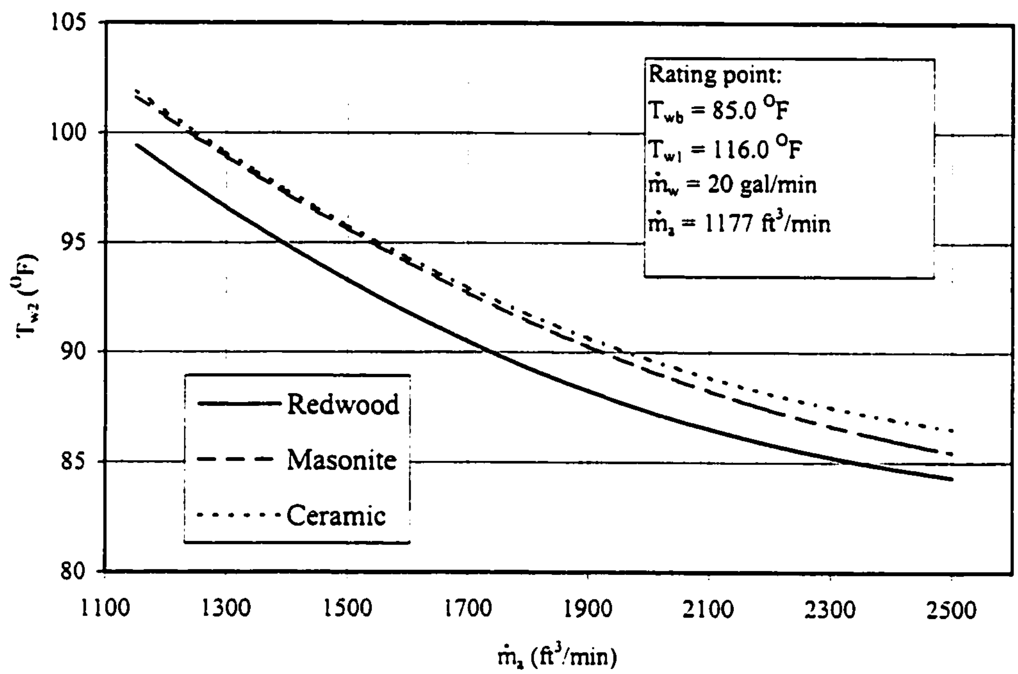

Figure 5.10: Effect of the mass flow rate of air on the outlet water temperature for the three different fill packed systems.

Figure 5.11 shows the variation of outlet water temperature with inlet water temperature for two different mass flow rates of water, whereas the wet-bulb temperature and the mass flow rate of air were kept constant. In the other figure (refer to Figure 5.12), the effect of wet-bulb temperature on the outlet water temperature is shown for the same mass flow rates of water. These parameters have the same effect on the outlet water temperature as discussed above, i. e., the effect of inlet water temperature is similar to that of the air wet-bulb temperature.

Another simulation is also carried out for an actual cooling tower. The data for this tower is also taken from CTI bulletin (1997). The tower is a counter flow splash fill type. The capacity of the tower is 9500 gal/min, and the mass flow rate of water-to-air ratio is 0.792. The design inlet and outlet water temperature is 104.7 and 79.3 ^{O}F respectively, with 73.1 ^{O}F wet bulb-temperature. These simulation are represented in Figure 5.13 and 5.14 for different values of the inlet hot water and wet-bulb temperatures, respectively.

5.3.2 Uncertainty analysis

In this section, sensitivity and uncertainty analysis results are presented for representative non-ideal conditions simulated on the cooling towers under consideration. The mathematical details of the analysis are discussed earlier in Section 4.4. The analysis is carried out from rating points of the cooling tower types mentioned earlier (R, M, and C tower types). The main objective of this analysis is to determine which parameter (independent variables: T_{w1}, T_{wb}, $\dot{m}_a$, $\dot{m}_w$) is affecting the results more, and how sensitive

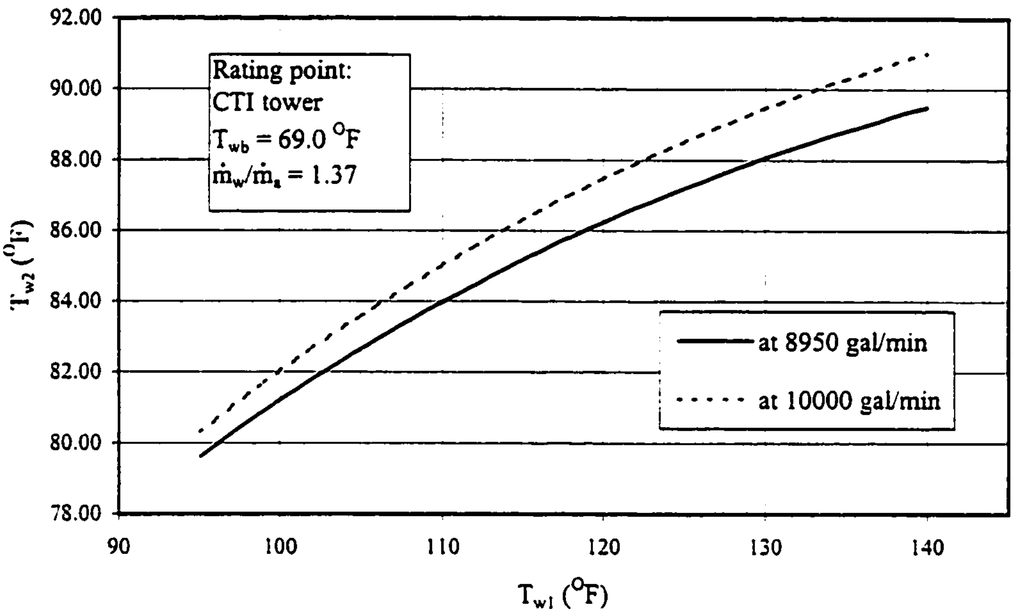

Figure 5.11: Effect of inlet water temperature on the outlet water temperature for two different mass flow rate of water, $\dot{m}_w/\dot{m}_a = 1.37$.

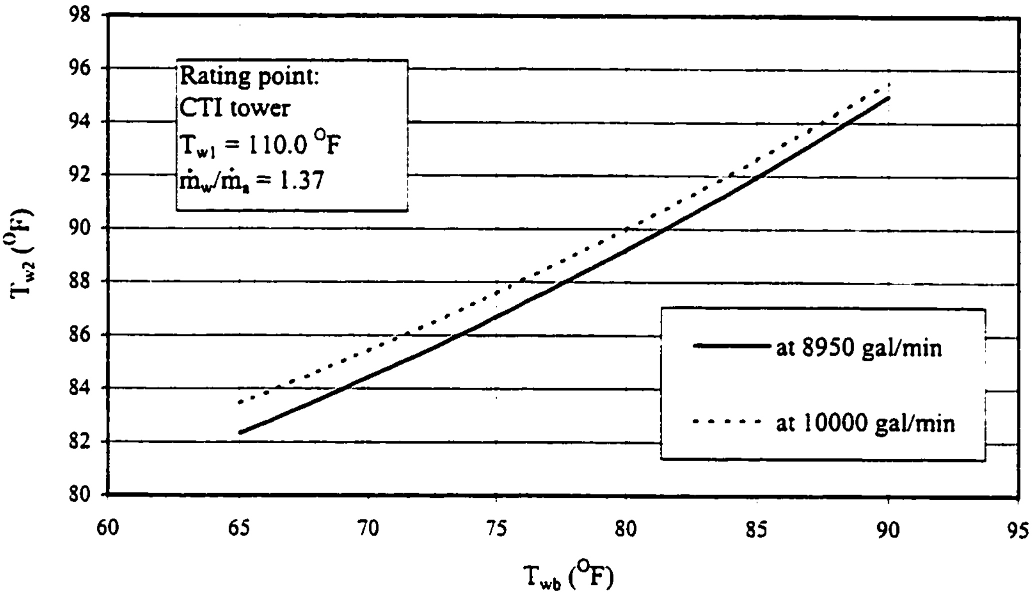

Figure 5.12: Effect of wet-bulb temperature on the outlet water temperature for two different mass flow rate of water, $\dot{m}_w/\dot{m}_a = 1.37$.

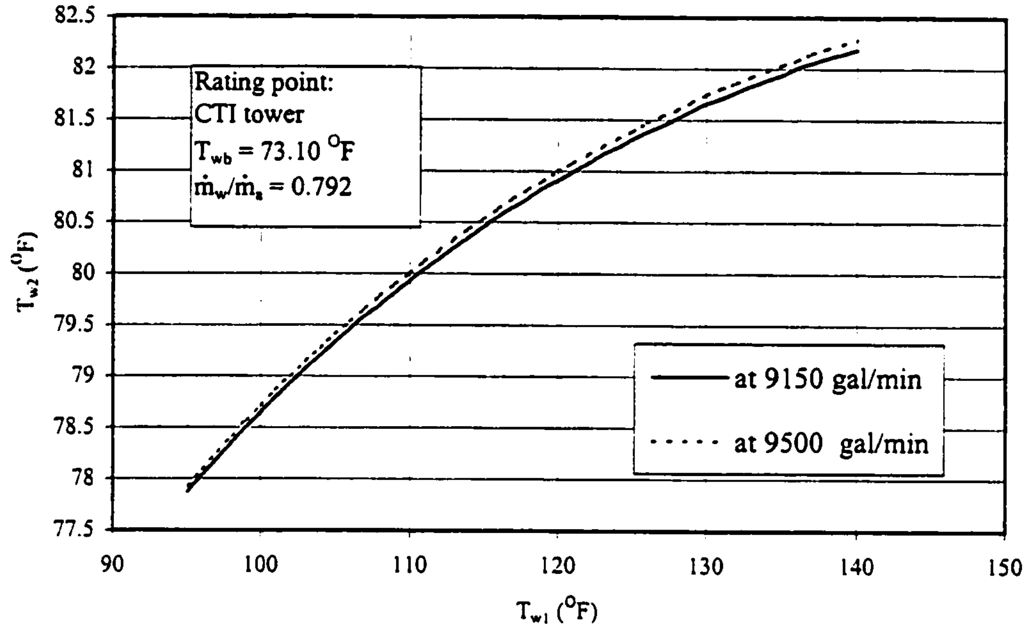

Figure 5.13: Effect of inlet water temperature on the outlet water temperature for two different mass flow rate of water, $\dot{m}_w/\dot{m}_a = 0.792$.

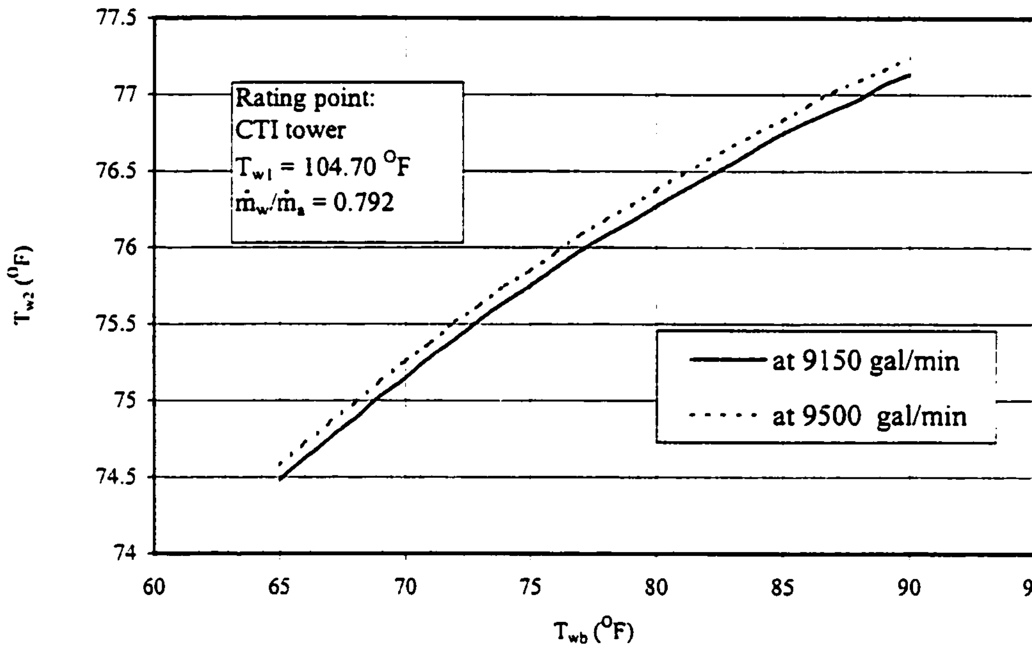

Figure 5.14: Effect of wet-bulb temperature on the outlet water temperature for two different mass flow rate of water, $\dot{m}_w/\dot{m}_a$ = 0.792.

the response to each one. Then, a comparison will be carried out for each parameter to the mentioned cooling tower fill packed types.

Table 5.2 shows a list of rating input variables, their corresponding nominal values and representative uncertainties (expressed as a fraction of their respective nominal values). The uncertainties in the inputs for inlet water temperatures and wet-bulb temperatures correspond to 0.5^0F (0.3^0C), and the uncertainties in the mass flow rates of water and air correspond to 3% (James et al., 1995).

Table 5.3 summarizes the nominal values of the input variables and their corresponding performance parameter for the three different fill types. Also, it summarizes the positive and negative perturbation values of the input variables and how can this perturbation affect the performance parameter. The nominal values of the performance parameter (T_{W2}) for the cooling tower types mentioned above and the corresponding overall uncertainties are shown in Table 5.4.

Tables 5.5, 5.6, and 5.7 show the values of the input variable uncertainties, Normalized Sensitivity Coefficient (NSC) and the relative contribution of each of the input variables to the overall uncertainty in the respective performance parameter. The results are also presented in the form of a bar diagram in Figure 5.15 to 5.20. The sign of sensitivity coefficients indicates the proportionality relation between the performance parameter and the input variable and its magnitude gives an indication of the strength of the relation. A positive value indicates direct proportion, and inverse proportion if otherwise. The mass flow rate of air m_a, for example, shows a negative value for its

Table 5.2: List of input variables, their corresponding nominal values and uncertainties considered for rating.

Input variables (X_i)	Nominal values (X)	Uncertainty ε_x
T_{wl} (^OF)	116	0.00431
T_{wb} (^OF)	85	0.00588
$\dot{m}_w$ (gal/min)	20	0.03
$\dot{m}_a$ (ft^3/min)	1177	0.03

Table 5.3: List of input variables, their corresponding nominal values and how can the positive and negative perturbation of each parameter affect the cooling performance.

i	T_{wl} (^{O}F)	T_{wb} (^{O}F)	$\dot{m}_w$ (gal/min)	$\dot{m}_a$ (ft^3/min)	Redwood T_{w2} (^{O}F)	Masonite T_{w2} (^{O}F)	Ceramic T_{w2} (^{O}F)
X_i	1177	20	85	116	100.2	102.45	101.51
T_{wl}+	1177	20	85	116.5	100.38	102.67	101.72
T_{wl}-	1177	20	85	115.5	100.01	102.22	101.31
T_{wb}+	1177	20	85.5	116	100.38	102.6	101.67
T_{wb}-	1177	20	84.5	116	100.02	102.3	101.36
$\dot{m}_w$+	1177	20.6	85	116	100.79	103.76	101.56
$\dot{m}_w$-	1177	19.4	85	116	99.58	101.06	101.48
$\dot{m}_a$+	1212.31	20	85	116	99.59	101.12	101.48
$\dot{m}_a$-	1141.69	20	85	116	100.8	103.78	101.56

Table 5.4: Nominal values of performance parameter in rating analysis for the three different fill packed materials.

	Nominal output (Y)	Normalized overall uncertainty (ε_Y)
Redwood T_{w2} (^{O}F)	100.2	0.00892
Masonite T_{w2} (^{O}F)	102.45	0.01869
Ceramic T_{w2} (^{O}F)	101.51	0.002592394

sensitivity coefficient indicating an inversely proportional relationship with the heat transfer process. This is because of the fact that as the air flow rate increases, the rate of heat transfer also increases.

5.3.3 Cooling Tower Packed With Different Fill System

It is seen that the input variables rank differently in regard to their sensitivity coefficient with respect to the performance parameter. With reference to Table 5.5 and Figure 5.15, the calculation of the outlet water temperature is most sensitive to the inlet water temperature, which is the only parameter to show sensitivity coefficient with magnitude of 0.43. However, as seen in Figure 5.18, the relative contribution of the uncertainty in the inlet water temperature to the overall uncertainty is relatively low. This is due to the comparatively low level of uncertainty in the temperature. Both of the mass flow rates of water and air show almost the same magnitude of sensitivity coefficient but with different proportional relationship with the outlet water temperature. The results show 0.2 and –0.2 for the mass flow rate of the water and air, respectively. These values led to have the same relative contribution of both parameters.

The results summarized in Table 5.6 and Figures 5.16 and 5.19 belong to the tower packed with masonite sheets. In general, the results behave the same as the cooling tower packed with redwood slats but with different numerical values. The magnitude of the normalized sensitivity coefficient for the inlet water temperature is the highest i. e, 0.51 (see Figure 5.16). As can be seen from Figure 5.19, the wet-bulb temperature contributing the least in the overall uncertainty.

Again the inlet water temperature appeared to be the most sensitive to the results for the tower packed with the ceramic rings with normalized sensitivity coefficient of 0.469, (see Figure 5.17 and Table 5.7). Also, the inlet water temperature is contributing the most in the overall uncertainty (see Figure 5.20). This due to the very low value of the normalized overall uncertainty (ε_Y).

It is important to mention that for all the fill pack systems studied in this thesis, the mass flow rates of water and air are equally sensitive to the results but in opposite way. For example, an increase in the mass flow rates of air by 3 % has the same effect as decreasing the mass flow rate of water by the same amount.

Table 5.5: Normalized sensitivity coefficient and relative contributions of input variables for the tower packed with redwood slats materials.

X_i	X	ε_x	NSC	R.C.
$T_{w1}(^{O}F)$	116	0.00431	0.428	0.0429
$T_{wb}(^{O}F)$	85	0.00588	0.305	0.0406
$\dot{m}_w$(gal/min)	20	0.03	0.201	0.4583
$\dot{m}_a$(ft^3/min)	1177	0.03	-0.201	0.4583

Table 5.6: Normalized sensitivity coefficient and relative contributions of input variables for the tower packed with Masonite sheets materials.

X_i	X	ε_X	NSC	R.C.
$T_{wi}(^{O}F)$	116	0.00431	0.510	0.01381
$T_{wb}(^{O}F)$	85	0.00588	0.249	0.00614
$\dot{m}_w$(gal/min)	20	0.03	0.439	0.49734
$\dot{m}_a(ft^3/min)$	1177	0.03	-0.433	0.48271

Table 5.7: Normalized sensitivity coefficient and relative contributions of input variables for the tower packed with ceramic ring materials.

X_i	X	ε_x	NSC	R.C.
$T_{wl}(^oF)$	116	0.00431	0.469	0.60686
$T_{wb}(^oF)$	85	0.00588	0.260	0.34693
$\dot{m}_w$(gal/min)	20	0.03	0.013	0.0231
$\dot{m}_a(ft^3/min)$	1177	0.03	-0.013	0.0231

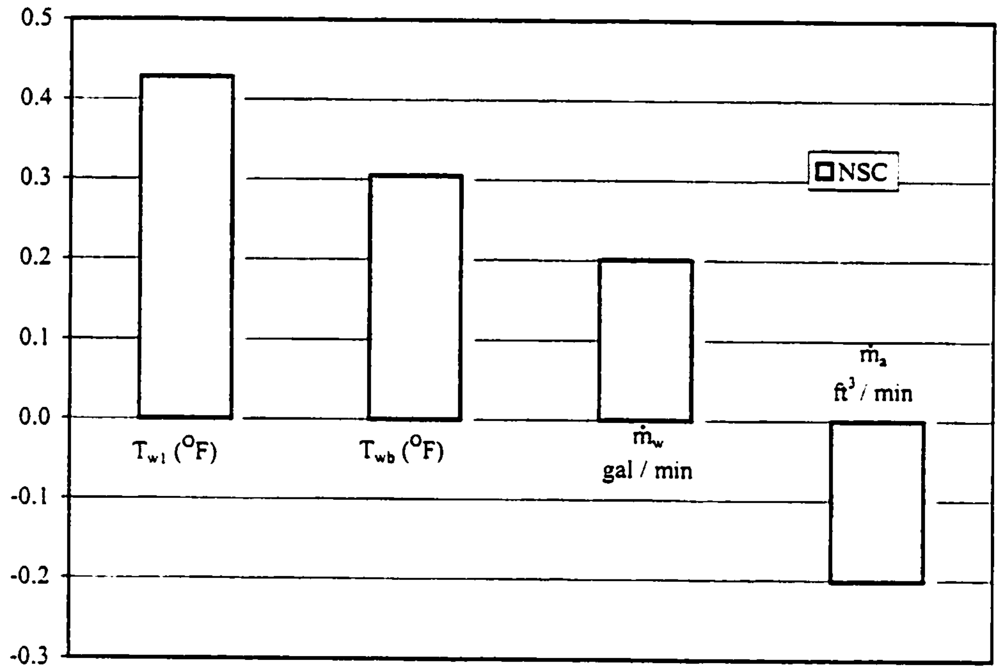

Figure 5.15: Normalized Sensitivity Coefficient (NSC) of the input variables for the tower packed with redwood slats.

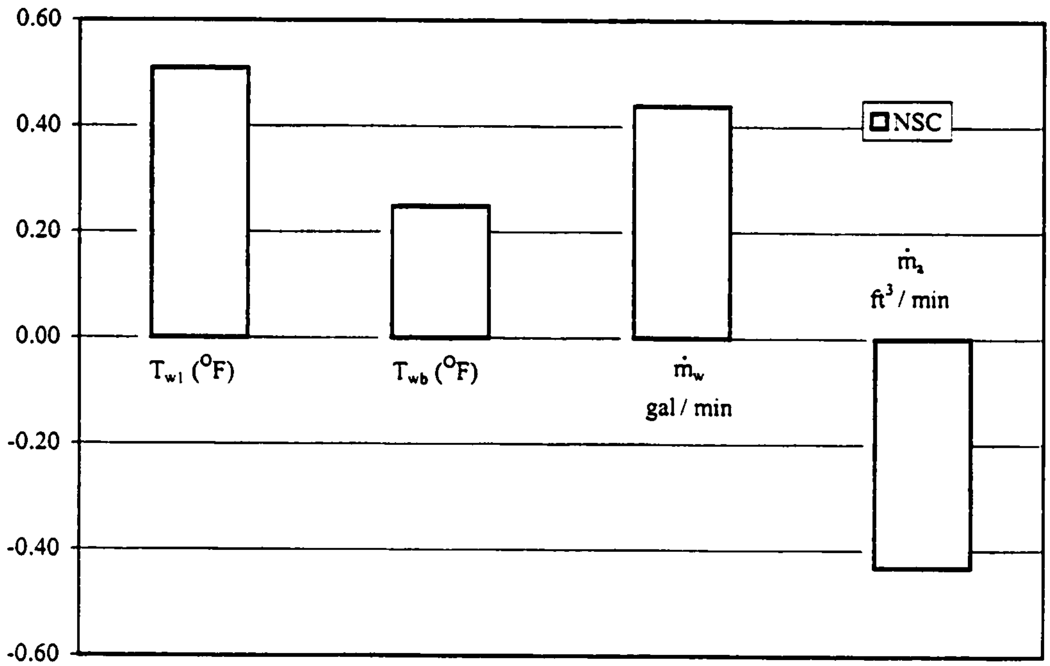

Figure 5.16: Normalized Sensitivity Coefficient (NSC) of the input variables for the tower packed with Masonite sheet.

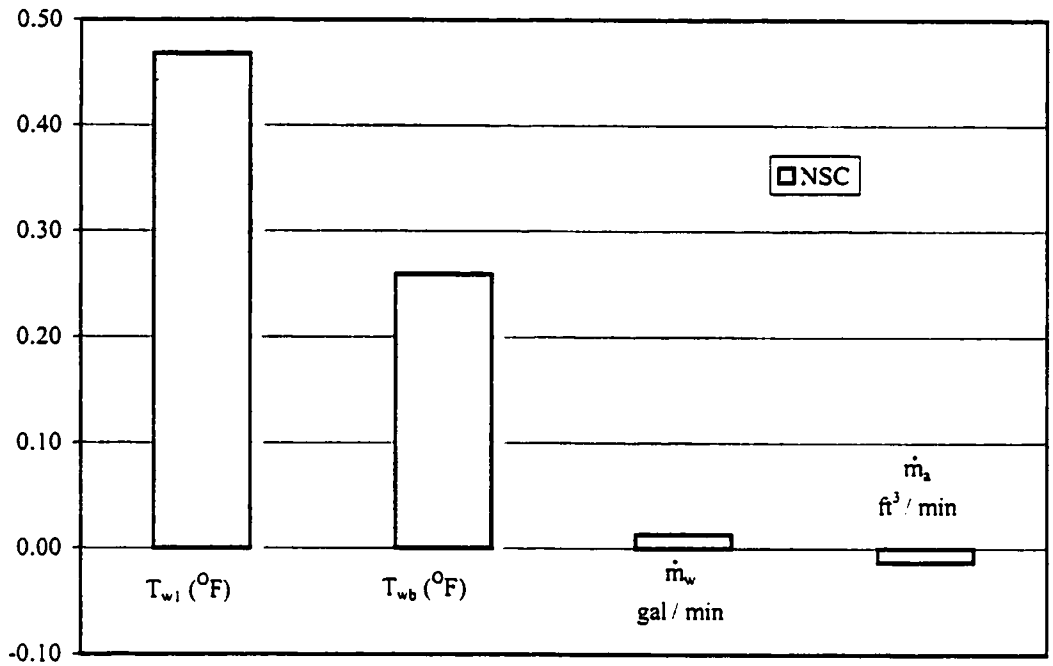

Figure 5.17: Normalized Sensitivity Coefficient (NSC) of the input variables for the tower packed with Ceramic rings.

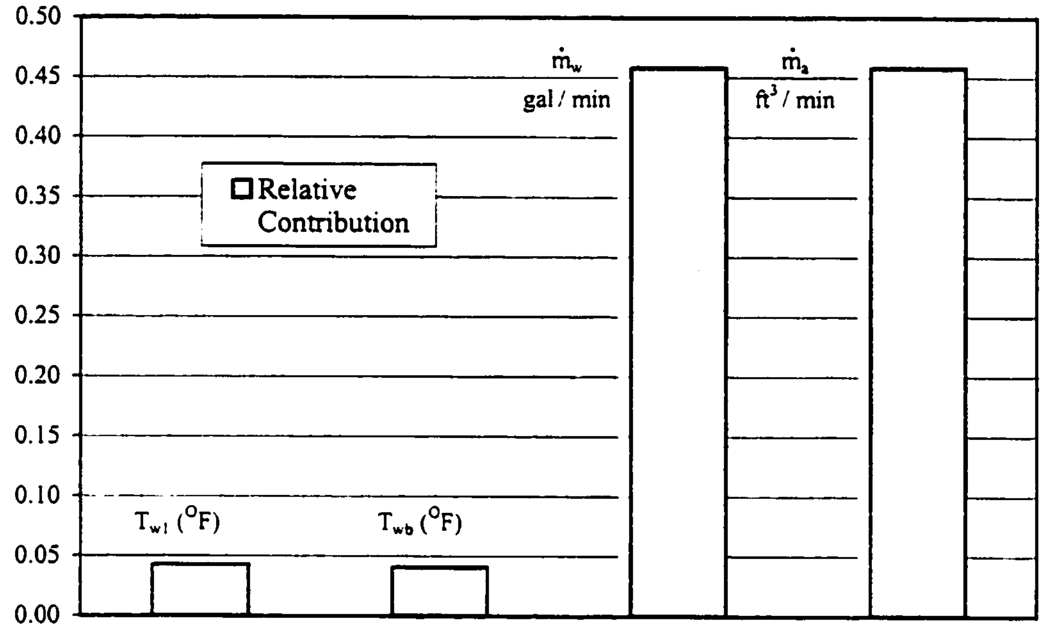

Figure 5.18: Relative Contribution (RC) of each of the input variables to the overall uncertainty in the tower packed with redwood slats.

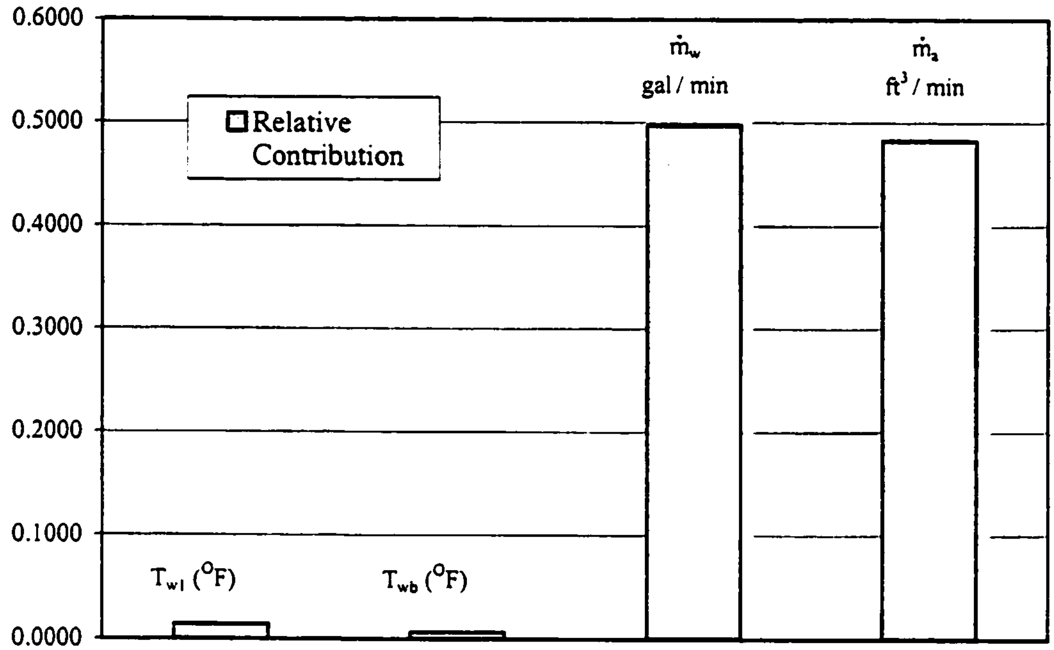

Figure 5.19: Relative Contribution (RC) of each of the input variables to the overall uncertainty in the tower packed with Masonite sheet.

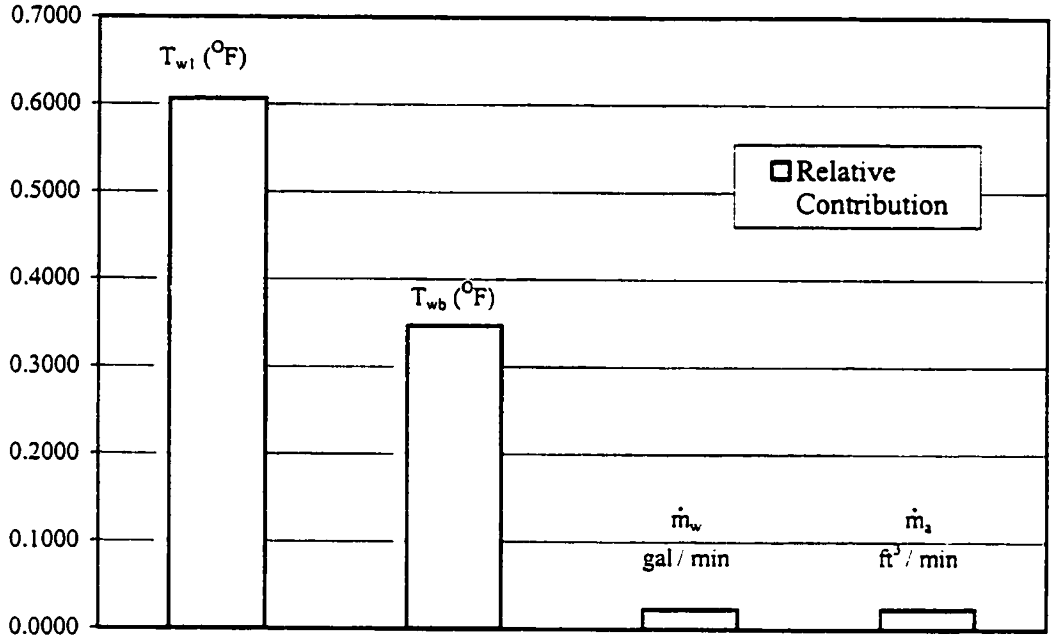

Figure 5.20: Relative Contribution (RC) of each of the input variables to the overall uncertainty in the tower packed with Ceramic rings.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

In this chapter, the conclusions drawn from the numerical study on the mechanical draught counter-flow cooling tower considered in this work are presented. Based on this research, a few recommendations for any general cooling tower thermal analysis are also given.

Conclusions

- The simulation results for a small cooling tower whose experimental data was available (Simpson and Sherwood, 1946) show that it is appropriate to consider the Merkel's assumptions. The maximum percentage error between the simulated and experimental values was 1.2%.
- From the parametric study it can be concluded that the wet-bulb temperature, the inlet water temperature, and the mass flow rate of water has a direct effect on the outlet water temperature. The mass flow rate of the air is the only parameter that has an inverse relationship with the outlet water temperature.

- It can be noted from the results that the tower packed with the Redwood slats material gives the best cooling performance because it provides the minimum outlet water temperature.
- A detailed sensitivity analysis shows that the inlet water temperature, out of all the parameter studied, has high Normalized Sensitivity Coefficient (NSC) to the performance results for all the fill packing systems investigated in this thesis.
- For the set of input variable uncertainties considered in this study, the uncertainties in the mass flow rates of air and water are found to be the primary contributors to the overall uncertainty in the outlet water temperature. The results have been obtained for a tower packed with Redwood slats and Masonite sheet. On the other hand, for the tower packed with ceramic rings, inlet water temperature and wet-bulb temperature has a higher relative contribution than the mass flow rates of air and water.

Recommendations

- There is not much experimental work regarding cooling tower performance analysis available in the literature. Therefore, it is recommended to carry out experiments either on a small cooling tower or an actual cooling tower.
- Similar to the performance results of Redwood slats, Masonite sheet, and Ceramic rings that are presented in this thesis, the effect of other types of fill pack systems such as cement block and film fill type on the cooling performance of the cooling towers can also be studied.

- A detail investigation on the limitations of Merkel's assumptions, particularly with regard to water loss due to evaporation could be of interest.

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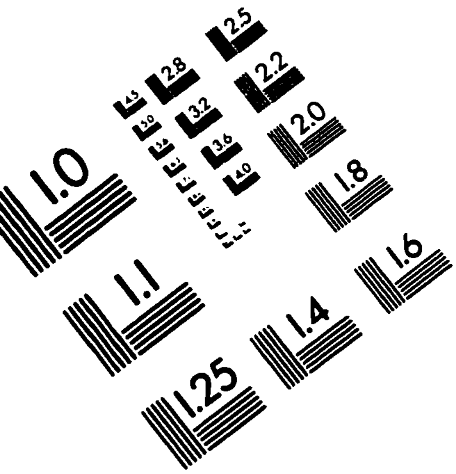
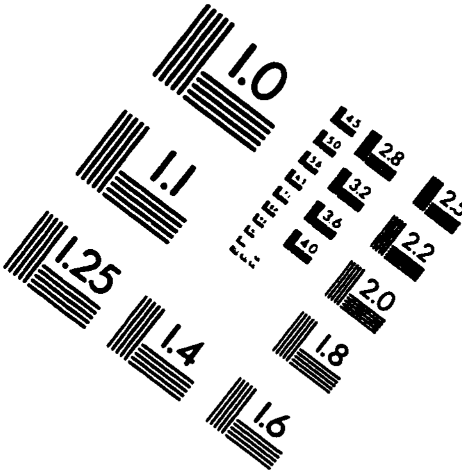
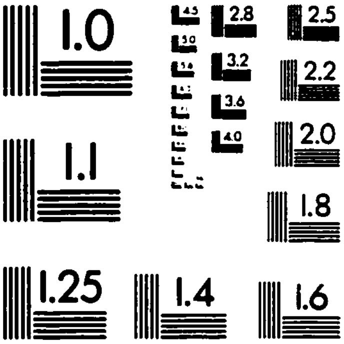
IMAGE EVALUATION TEST TARGET (QA-3)







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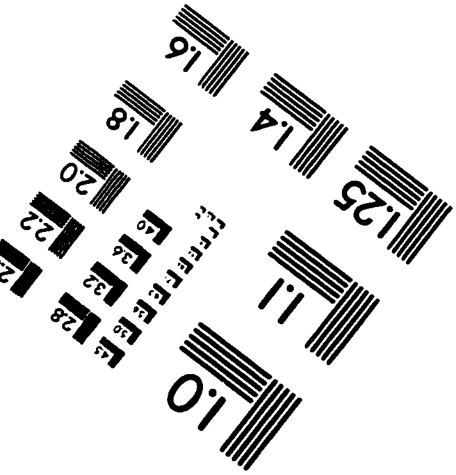

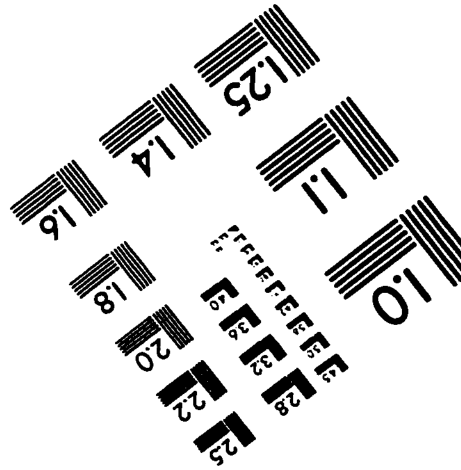

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